

CRANFIELD UNIVERSITY

AJIMOTOKAN, HABEEB A.

**A STUDY OF TRILATERAL FLASH CYCLES FOR  
LOW-GRADE WASTE HEAT RECOVERY-TO-  
POWER GENERATION**

SCHOOL OF ENGINEERING  
Energy and Power Engineering Division,  
Cranfield University, Cranfield, Bedfordshire MK43 0AL, UK

PhD  
Academic Year: 2013 - 2014

Supervisors: Dr. I. Sher  
& Prof. H. Yeung

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## ABSTRACT

There has been renewed significance for innovative energy conversion technologies, particularly the heat recovery-to-power technologies for sustainable power generation from renewable energies and waste heat. This is due to the increasing concern over high demand for electricity, energy shortage, global warming and thermal pollution. Among the innovative heat recovery-to-power technologies, the proposed trilateral flash cycle (TFC) is a promising option, which presents a great potential for development. Unlike the Rankine cycles, the TFC starts the working fluid expansion from the saturated liquid condition rather than the saturated, superheated or supercritical vapour phase, bypassing the isothermal boiling phase. The challenges associated with the need to establish system design basis and facilitate system configuration design-supporting analysis from proof-of-concept towards a market-ready TFC technology are significant. Thus, there is a great need for research to improve the understanding of its operation, behaviour and performance. The objective of this study is to develop and establish simulation tools of the TFCs for improving the understanding of their operation, physics of performance metrics and to evaluate novel system configurations for low-grade heat recovery-to-power generation. This study examined modelling and process simulation of the TFC engines in order to evaluate their performance metrics, predictions for guiding system design and parameters estimations. A detailed thermodynamic analysis, performance optimization and parametric analysis of the cycles were conducted, and their optimized performance metrics compared. These were aimed at evaluating the effects of the key parameters on system performances and to improve the understanding of the performance behaviour. Four distinct system configurations of the TFC, comprising the simple TFC, TFC with IHE, reheat TFC and TFC with feed fluid-heating (or regenerative TFC) were examined. Steady-state steady-flow models of the TFC power plants, corresponding to their thermodynamic processes were thermodynamically modelled and implemented using engineering equation solver (ESS). These models were used to determine the optimum synthesis/ design parameters of

the cycles and to evaluate their performance metrics, at the subcritical operating conditions and design criteria. Thus, they can be valuable tools in the preliminary prototype system design of the power plants. The results depict that the thermal efficiencies of the simple TFC, TFC with IHE, reheat TFC and regenerative TFC employing *n*-pentane are 11.85 - 21.97%, 12.32 - 23.91%, 11.86 - 22.07% and 12.01 - 22.9% respectively over the cycle high temperature limit of 393 - 473 K. These suggest that the integration of an IHE, fluid-feed heating and reheating in optimized design of the TFC engine enhanced the heat exchange efficiencies and system performances. The effects of varying the expander inlet pressure at the cycle high temperature and expander isentropic efficiency on performance metrics of the cycles were significant. They have assisted in selecting the optimum-operating limits for the maximum performance metrics. The thermal efficiencies of all the cycles increased as the inlet pressures increased from 2 - 3 MPa and increased as the expander isentropic efficiencies increased from 50 - 100%, while their exergy efficiencies increased. This is due to increased net work outputs that suggest optimal value of pressure ratios between the expander inlets and their outlets. A comprehensive evaluation depicted that the TFC with IHE attained the best performance metrics among the cycles. This is followed by the regenerative TFC whereas the simple TFC and reheat TFC have the lowest at the same subcritical operating conditions. The results presented show that the performance metrics of the cycles depend on the system configuration, and the operating conditions of the cycles, heat source and heat sink. The results also illustrate how system configuration design and sizing might be altered for improved performance and experimental measurements for preliminary prototype development.

**Keywords:**

Waste heat recovery-to-power, power cycle, trilateral flash cycle, process integration, modelling, process simulation, energy analysis, exergy analysis, performance optimization, parametric analysis, performance comparison.

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## **LIST OF ABBREVIATIONS**

ESS    Engineering equation solver

HP     High pressure

ICE    Internal combustion engine

LP     Low pressure

ORC   Organic Rankine cycle

SSSF   Steady-state steady-flow

TFC   Trilateral flash cycle

## NOMENCLATURE

1, 2, 3, 4      States of working fluid

$E$       Energy (J)

$Ex$       Exergy (J)

$h$       Specific enthalpy (J kg<sup>-1</sup>)

$P$       Pressure (Pa)

$S$       Entropy (J Kg<sup>-1</sup>K<sup>-1</sup>)

$T$       Temperature (K)

$V$       Total volume (m<sup>3</sup>)

$v$       Specific volume (m<sup>3</sup> kg<sup>-1</sup>)

$\dot{E}$       Energy rate (J kg<sup>-1</sup>)

$\dot{Ex}$       Exergy rate (J kg<sup>-1</sup>)

$\dot{m}$       Mass flow rate (kg s<sup>-1</sup>)

$\dot{Q}$       Heat transfer rate (J kg<sup>-1</sup>)

$\dot{W}$       Specific work (J kg<sup>-1</sup>)

### Greek letters

$\Delta$       Change

$\eta$       Efficiency (%)

### Subscripts

$o$       Reference state

$c$       Cold stream/ heat sink

$con$       Condenser/ condensing

$cp$       Cooling process

$cs$       Cooling system

$cyc$       Cycle

$dest$       destroyed

exp	Expander
gen	Generation
h	Hot stream/ heat source
hp	High pressure/ heating process
hx	Heat exchanger
in	In/ Inlet
is	Isentropic
lp	Low pressure
max	Maximum
opt	Optimum
out	Out/ Outlet
p	Pump
rev	Reversible
rh	Reheating
sys	System
th	Thermal
u	Useful
wf	Working fluid

### **Superscripts**

in	In/ Inlet
out	Out/ Outlet



# **CHAPTER ONE**

## **INTRODUCTION**

# 1 Introduction

The need for innovative energy conversion technologies is significant for sustainable power generation. This is due to the growing concern over the Earth's limited fossil fuel reserves (Smith, 1993), rising cost of energy prices in the global market (Sanjay, 2011), high demand for electricity by an increasing world population, energy shortage, effects of greenhouse gas emissions (Quoilin et al., 2010; Khennich and Galanis, 2012) and thermal pollution (Dai et al., 2009b).

Furthermore, because the primary energy resources and their conversion systems play a significant role in the world economy, repeated efforts have been made to study them (Miketa and Mulder, 2005; Valente, 2005) to improve fuel economy and efficiency, and significantly reduce primary energy demand and emissions (Patil et al., 2009; Fang et al., 2013). This is definitely not just a method to conserve these limited resources and the environment but also to provide a sustainable and affordable power supply.

Among these energy conversion technologies, there has been renewed significance for heat recovery-to-power technologies. This is due to the growing interest in heat recovery-to-power generation from renewable energies such as solar thermal, geothermal and biomass; and waste heat from the conventional fossil fuel energy systems. A number of novel heat recovery-to-power technologies have been proposed for heat recovery-to-power generation from low-to-medium temperature heat sources. Among the proposed innovative

solutions, the trilateral flash cycle (TFC) presents a great potential for development.

The concern to attain maximum reversible work from a power cycle using a non-isothermal single-phase heat sources has led to the concept of the TFC, a modified power cycle for heat recovery-to-power applications that Smith in the 19th century had figured out (Smith, 1993). Smith (1993) proposed and described the cycle as one of the optimum alternatives among the heat recovery-to-power technologies capable of attaining this ideal. This cycle also known as the trilateral cycle (TLC) or trilateral wet vapour cycle is a power (or thermodynamic) cycle at its basis proposed to enhance the performance of heat recovery-to-power especially from non-isothermal low-to-medium temperature heat sources (Brown and Mines, 1998).

Basically, the TFC engine has the same components as the Rankine-cycle engines but unlike the Rankine cycle, it does not evaporate the working fluid during the heating phase; instead expands it, from the saturated liquid condition, as a two-phase mixture. This feat permits better thermal matching during the heat transfer from the heat source or non-isothermal heat source (i.e. of variable temperature) to the working fluid (Chan et al., 2013), minimising exergy destruction and improving performance metrics.

Furthermore, there is a growing interest in this cycle because it thermally matches the exergy of the temperature profiles of non-isothermal heat source and functions at rational pressures such that its application is viable economically for mechanical or electrical power production (Zamfirescu and

Dincer, 2008). It could attain as nearly as possible the theoretical limit of performance at moderate expansion machine inlet temperature and pressure, and is characterized by high power density.

Heat recovery-to-power technology as an overall efficiency improvement measure, unlike renewable sources, is one renewable approach where waste heat is used as energy source for sustainable power generation (Van Der Linden and Romero, 2009). Moreover, the distributed generation is one of the numerous merits of waste heat recovery-to-power generation, by generating power at its point of use, thus eliminating 4 - 8% losses in power during transmission and distribution as related to central generation (Van Der Linden and Romero, 2009).

There are two usual arrangements (or techniques) of the heat recovery-to-power schemes. This include the recuperation, where the waste heat recovered from the same cycle is employed to enhance the cycle efficiency; and the bottoming cycles, where the waste heat is employed as a source of heat for an independent cycle (Heppenstall, 1998; Franco and Casarosa, 2002) such as the steam Rankine cycle, organic Rankine cycle, supercritical Rankine cycle and TFC.

The choice of a suitable working fluid for a given application, the feasible operating conditions/ parameters and the system configuration design, are the most essential features of any thermal system design (Dincer and Al-Muslim, 2001; Chen et al., 2011a; Sahin et al., 2013). With the application of a suitable high molecular working fluid, the TFC power plants could recover and convert

heat energy efficiently from renewable thermal sources as well as from the by-product (waste heat) of numerous non-renewable sources into mechanical or electrical power. The high molecular mass allows efficient exploitation of non-isothermal single-phase heat sources for heat recovery-to-power generation, which allows the net work output of the TFC in a wide range of power capacities (Invernizzi et al., 2007; Baral and Kim, 2014). It could span for a unit, from a few kilowatts to a few megawatts; while poly-units in parallel, could form a multi-megawatt in numerous applications, such as geothermal, solar thermal, biomass, ocean thermal and desalination power plants among others.

The by-products (waste heat) of the combustion from an internal combustion engine that are dissipated into the environment pose severe environmental concerns. From the perspective of thermal equilibrium, merely 26 - 34% of the total thermal energy produced as a result of combustion in internal combustion engine accounts for the output power driving the engine (Dincer and Al-Muslim, 2001). While numerous studies have estimated that as much as 20 - 50% of the primary energy consumed industrially is exhausted as waste heat (Fang et al., 2013; Hung et al., 1997; Johnson et al., 2008; Lian et al., 2011). Subsequently, about 70% of the entire combustion heat of engines and about 50% of the entire primary energy consumed industrially are dissipated in the environment, demonstrating the vast potential for waste heat recovery-to-power generation.

## **1.1 Aim and Objectives**

The aim of this study is to develop and establish simulation tools of the TFCs for improving the understanding of their operation, physics of performance metrics

and to evaluate novel system configurations for low-grade waste heat recovery-to-power generation.

The research specific objectives are to:

1. Identify the problems and potential of the TFC systems for low-grade waste heat recovery-to-power generation;
2. Develop a modelling approach to simulate the TFC systems;
3. Analyse the basic processes involved in the TFC systems for low-grade waste heat recovery-to-power generation;
4. Evaluate innovative improvement schemes of the TFCs to improve heat exchange performance and system efficiency; and
5. Establish a simulation-based rigorous performance study of TFC systems for low-grade waste heat recovery-to-power generation.

## **1.2 Motivation**

About 70% of primary energy is discharged to the environment as waste heat in exhaust gases from conventional fossil fuel energy systems (internal combustion engines and gas turbines) as well as about 30% from combined-cycle systems, and about 50% of primary energy consumed industrially, resulting in severe environmental (thermal) pollution.

Concerns over the environmental challenges, coupled with the socioeconomic pressures and the vast availabilities of low-grade heat streams of many renewable energy sources like the solar thermal, geothermal and biomass; and vast waste heat generation have driven the development of numerous waste

heat recovery-to-power cycles and several novel concepts that have been proposed.

Among these proposed cycles, the TFC for heat recovery-to-power generation is, no doubt, a promising option to satisfy the numerous requirements in electricity demand, environment and performance. The fundamental principles that govern its processes might be understood, but the cycle applications require innovative solutions to challenges experienced from working fluid selection through plant design, development, operation, physics of behaviour and maintenance.

One of the challenges is that a wide spectrum of low-grade waste streams is unexploited. The need to examine available heat sources and the power cycle, in order to craft the power plant to match the exergy of the heat source profile is significant. Moreover, how to exploit as many as possible of these numerous energy sources, which are by their nature intermittent, onto the grid as well as for off-grid electricity supply without degrading the reliability of supply is challenging.

As observed with other alternative energy technologies like the solar; development, innovation and investment would drive growth after a market-ready technology is attained. Whether it be the efficient solar cell or the large wind turbine, the future prospect of the TFC power plants depend on the comparative improvements. This can be achieved especially through system modelling, process optimization and simulation, and analyses to improve system efficiency as well as efficiencies of development undertakings and

reduce costs. Thus, the proposed TFC technology as well as its variants based on the optimized system design is studied for low-grade heat recovery-to-power generation.

### **1.3 Research Scope**

This study directs its attention to establishing and implementing simulation tools for TFCs to improve the understanding of their operation and physics of performance, and to evaluate novel configurations for low-grade waste heat recovery-to-power generation.

The study commenced with the assessment of the technologies associated with heat recovery-to-power generation. Afterwards, the system design criteria were established subject to engineering design constraints and the different components of the TFCs corresponding to their thermodynamic processes were thermodynamically modelled and implemented, employing EES (engineering equation solver). The EES is a commercially available software with in-built thermodynamics and transport properties for working fluids (Klein, 2013). This software enables the introduction and modification of component models, allowing consistent analyses of the process and conceptual system design thermodynamically. It was also employed for the performance optimization of the cycle to maximise the cycle net work output and/ or thermal efficiency.

In order to model the systems, a semi-empirical modelling approach was adopted, which requires a limited number of meaningful parameters. These parameters, which are easily distinguished for process and conceptual system



configuration design, were established subject to engineering design constraints, manufacturers' manual and the open literature.

Thermodynamic and parametric analyses of the TFC systems for low-grade heat recovery-to-power generation were performed as a solution to evaluate the performance of the cycles and their predictions for guiding conceptual design, sizing and parameters estimations. Four distinct TFC power plants were thermodynamically and parametrically analysed. Performance optimization and future heat exchange improvement scheme of the optimized system design of the TFCs were examined.

Lastly, performance analysis and comparative study of the TFCs for low-grade heat recovery-to-power generation were performed.

## **1.4 Expected Findings**

The novelty and contributions to knowledge are:

1. Establishment of a modelling approach and simulation tools for TFC systems;
2. Implementation of the simulation tools for understanding the requirements to facilitate a procedure to system configuration design and system development;
3. Implementation of innovative heat exchange and system performance improvement schemes of the optimized design of the TFC; and
4. Determination of key operating parameters through sensitivity studies to establish system configuration design basis for TFC power plants (which

does not include off-design operations), and to understand the interrelationship among input parameters and system performance metrics.

## 1.5 Definitions of Terms

**Bottoming cycle** is a power cycle operating at lower average temperatures that receives heat from a power cycle operating at higher average temperatures (Cengel and Boles, 2006).

**Cycle** is a process, or series of processes, that allows a system to undergo state changes and returns the system to the initial state at the end of the process. That is, for a cycle the initial and final states are identical (Cengel and Boles, 2006).

**Efficiency** is defined as the 'ratio of desired result for an event to the input required to accomplish the event'. Efficiency is one of the most frequently used terms in thermodynamics, and it indicates how well an energy conversion or transfer process is accomplished (Cengel and Boles, 2006).

**Exergy** can be defined as 'the maximum amount of work produced by a flow of matter or energy in a process or system as it brings to equilibrium with a reference reservoir or its environment' (Joshi et al., 2009).

**Exergy analysis** can be expressed as 'a method which employs the principles of conservation of mass and energy coupled with the second law of thermodynamics for the analysis, design and improvement of process of a system, particularly energy systems' (Dincer and Rosen, 2007).

**Expander** is a device that 'produces shaft work due to a decrease of enthalpy, kinetic, and potential energies of a flowing fluid' (Cengel and Boles, 2006).

**Heat engines** are devices designed for the purpose of converting other forms of energy (typically in the form of heat) to work (Cengel and Boles, 2006).

**Heat source** is a heat reservoir that 'supplies energy in the form of heat' while the **heat sink** is a heat reservoir that 'absorbs energy in the form of heat' (Cengel and Boles, 2006).

**Irreversibilities** are the factors that 'cause a process to be irreversible'. They include friction, unrestrained expansion, mixing of two gases, heat transfer across a non-isothermal finite temperature difference, electric resistance, inelastic deformation of solids, and chemical reactions (Cengel and Boles, 2006).

**Process** is any change that a system undergoes from one equilibrium state to another. To describe a process completely, one should specify the initial and final states of the process, as well as the path it follows, and the interactions with the surroundings (Cengel and Boles, 2006).

**State** is the condition of a system not undergoing any change, which gives a set of properties that completely describes the condition of that system. At this point, all the properties can be measured or calculated throughout the entire system (Cengel and Boles, 2006).

**Steady-flow process** is a process during which a fluid flows through a control volume steadily. That is, the fluid properties can change from point to point

within the control volume, but at any point, they remain constant during the entire process. During a steady-flow process, no intensive or extensive properties within the control volume change with time (Cengel and Boles, 2006).

**Topping cycle** is a power cycle operating at the high average temperature that rejects heat to a power cycle operating at the lower average temperature (Cengel and Boles, 2006).

**Working fluid** is the fluid to and from which heat and work is transferred while undergoing a cycle in heat engines and other cyclic devices (Cengel and Boles, 2006).

## **1.6 Thesis Structure**

Pathways from proof-of-concept towards a market-ready technology of the TFC system as well as its proposed innovative improvement schemes on the basis of a modified (or optimized) design for waste heat recovery-to-power generation are studied and reported in this thesis. Each of its chapters comprises an element of the bigger picture, consisting of three journal papers and one conference proceeding. These papers are either accepted, pending acceptance or being prepared for publication. The thesis is outlined such that all the elements of this research study are briefly summarised as follows.

**CHAPTER ONE** gives an overview of the background of study, its motivation and scope as well as the definitions of terms, thesis structure and research accomplishments to date.

In **CHAPTER TWO**, a comprehensive review of literature is provided. This includes the definition of low-grade heat, description of the numerous major sources of low-grade heat that are under-exploited, as well as the discussions of the major heat recovery-to-power technologies, working fluid and its selection criteria, and a systemic review of innovative heat recovery-to-power engines among others. This chapter provides the basis for this study and how system configuration design and sizing might be altered for improved performance.

**CHAPTER THREE** examines the modelling, simulation and evaluation of the performance parameters of the TFC engines for low-grade waste heat recovery-to-power generation. The simple TFC and proposed novel improvement schemes of the optimized design of the TFC for improved heat exchange and system performances are studied. For the proposed cycles, the fixed values of the input parameters and key decision variables are defined, and system configuration design criteria are established subject to engineering design constraints or the defined fixed parameters. Based on the theory of steady-state steady-flow thermodynamics, the models of the different components of the TFC systems corresponding to their thermodynamic processes are established and implemented. The simple TFC, which is the classical design, is expected to be the reference engine.

In **CHAPTER FOUR**, the thermodynamic solutions of the TFCs operating at the subcritical conditions with low-grade heat in the temperature limit of 393 - 473 K for low-grade waste heat recovery-to-power generation are examined. This is to assess the performance metrics of the proposed cycles and their predictions. Four different system configuration designs of the TFC with distinct working

principles, which include the simple TFC, TFC with internal heat exchanger, reheat TFC and TFC with feed fluid-heating (known as regenerative TFC) are studied for improved system performance.

In **CHAPTER FIVE**, the performance optimization and parametric solutions of the TFC systems operating at the subcritical conditions with low-grade heat in the temperature limit of 393 - 473 K for heat recovery-to-power generation are examined. The parametric sensitivity studies of the models with the key parameters (variables): the expander inlet pressure at the cycle high temperature and expander isentropic efficiency are examined by standard process simulation experimentations of variation of pressure at the cycle high temperature and expander isentropic efficiency. Results include the cycles' performances and some sensitivity studies results that illustrate the effects of key parameters on the thermodynamic performances of the cycles.

**CHAPTER SIX** provides the optimized performance analysis and comparison of the TFCs for low-grade heat recovery-to-power generation. The combined results of the cycles' thermodynamic performances, thermodynamic analyses, and performance optimization and sensitivity studies obtained in CHAPTERS THREE, FOUR and FIVE respectively are discussed in a wider perspective.

Lastly, conclusions are drawn from the research outcomes as well as recommendations for future work to further progress substantially and optimise the proposed TFCs for low-grade heat recovery-to-power is provided in **CHAPTER SEVEN**.

## 1.7 Research Accomplishments

1. **Ajimotokan, H.A.**, Sher, I., Biliyok, C. and Yeung, H. **(2014)** 'Trilateral Flash Cycle for Recovery of Power from a Finite Low-Grade Heat Source'. ***Computer Aided Chemical Engineering*, 33; 1831-1836.** Published by Elsevier, the Netherlands.
2. **Ajimotokan, H.A.**, Sher, I. and Yeung, H. **(2014)** 'Modelling and Simulation of Performance Parameters of a Trilateral Flash Cycle System'. ***Conference Poster Session at the 2014 Engineering & Aerospace Doctoral Training Centre's Research Students' and Supervisors' Poster Conference***, School of Engineering, Cranfield University, Cranfield, UK; 4<sup>th</sup> March, 2014.
3. **Ajimotokan, H.A.**, Sher, I. and Yeung, H. **(2013)** 'The Study of a Trilateral Flash Cycle for Conversion of Low-Grade Heat into Power'. ***Conference Poster Session at the 2013 Engineering & Aerospace Doctoral Training Centre's Research Students' and Supervisors' Poster Conference***, School of Engineering, Cranfield University, Cranfield, UK; 12<sup>th</sup> February, 2013.
4. **Ajimotokan, H.A.** and Sher, I. 'Trilateral Flash Cycles for Low-Grade Waste Heat Recovery-to-Power Generation - I. Modelling, Simulation and Evaluation'. *Pending Acceptance*
5. **Ajimotokan, H.A.** and Sher, I. 'Trilateral Flash Cycles for Low-Grade Waste Heat Recovery-to-Power Generation - II. Thermodynamic Analysis'. *Pending Acceptance*

6. **Ajimotokan, H.A.** and Sher, I. 'Trilateral Flash Cycles for Low-Grade Waste Heat Recovery-to-Power Generation - III. Performance Optimization and Parametric Analysis'. *Being Prepared for Publication*
7. **Ajimotokan, H.A.** and Sher, I. 'Optimized Performance Analysis and Comparison of Trilateral Flash Cycles for Low-Grade Waste Heat Recovery-to-Power Generation'. *Being Prepared for Publication*



## **CHAPTER TWO**

### **LITERATURE REVIEW**

## **2 Background**

A variety of the renewable energies and exhaust waste of low-grade heat sources that can be employed for power generation or useful mechanical (or shaft) work are highlighted, reviews of the major power cycle for low-grade heat recovery-to-power generation are discussed, working fluid and its selection criteria, expanders for low-grade heat recovery-to-power generation system and energy related terms are discussed as well as a systemic review of available published works in the literature for heat recovery-to-power technologies.

### **2.1 Low-Grade Heat**

Low-grade (or low-to-medium temperature) heat is essentially low- and mid-temperature heat with less exergy density, which cannot be converted efficiently by conventional methods. Though, there is no definite specification on the temperature range of low-grade heat, it is acknowledged that a heat source with temperature limit between 70 - 370°C is considered as a low-grade heat source (Mathias et al., 2009; Chen, 2010; Chen, 2011a).

### **2.2 Low-Grade Heat Sources**

Typically, one of the products of the combustion of hydrocarbon fuels is the high-temperature energy (or heat) source. Though, as the temperature of the heat source diminishes, numerous renewable energies and waste heat sources are examples of low-to-medium temperature energy (or heat) source.

The major low-grade heat sources are either from renewable energy sources like solar thermal, geothermal and biomass; or from exhaust waste heat from

conventional fossil fuel energy systems like internal combustion engines, micro-gas turbines, gas turbines, steam turbines and combined-cycle systems, or industrial waste heat.

### **2.2.1 Solar Thermal**

The solar radiation is a high-temperature, high-exergy energy source at its origin, the Sun, whose irradiance is about  $63 \text{ MW/m}^2$  and which travels to the earth in the form of electromagnetic radiation. Although, the solar thermal energy arriving the Earth surface decreases dramatically to about  $1 \text{ kW/m}^2$ , due to the Sun-Earth separation geometry (Lasode and Ajimotokan, 2011). This low energy density and the source availability fluctuations are the key drawbacks in solar thermal energy exploitations (Baral and Kim, 2014). The uses of optical concentrating solar systems that transform the concentrated radiation typically into solar thermal energy overcome this deficiency (Romero et al., 2004; Fernandez-Garcya et al., 2010; Romeo et al., 2011). The commercially available technologies, photovoltaic modules, which are used for direct conversion of solar thermal energy into electrical energy; have low efficiency and batteries are required for energy storage (Baral and Kim, 2014).

The solar radiation incident on the surface of the Earth is at a rate of about  $1.7 \times 10^{17}$  watts (Goswami et al., 2000), a value 10,000 times, the present world energy consumption (National Energy Information Center, 2010). In order to make this enormous energy source more useable, optical concentrating solar systems like solar thermal collectors, solar ponds and so on are used to collect and convert the solar radiation into solar thermal energy (Chen, 2010).

Solar thermal collectors are classified as low-temperature, medium-temperature and high-temperature collectors based on the collecting temperatures. Typically, the low-temperature collectors are flat-plate collectors with (or without) glazing, which comprise an absorber surface with thermal insulation behind, as well as a trap to diminish energy losses and a heat transfer medium (Chen, 2010). While the medium- and high-temperature collectors are made using double-glazed flat-plate collectors, evacuated tube collectors and concentrating collectors with (or without) tracking device (Chen, 2010; Goswami et al., 2000).

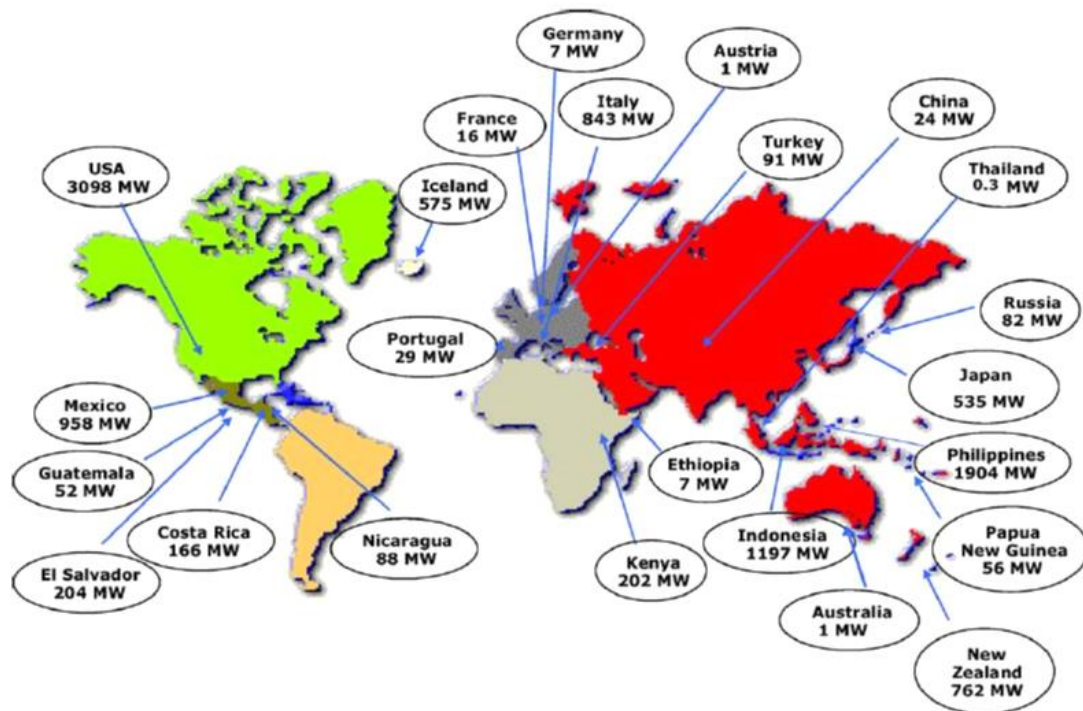
On the other hand, the solar ponds, which are salt water pools with a bottom-to-top density gradient; are the large-scale solar thermal collectors that combine heat collection and storage (Chen, 2010). Figure 2.1 is a research pond jointly built in Pyramid Hill, Victoria, Australia by RMIT, Geo-Eng Australia Pty Ltd. and Pyramid Salt Pty Ltd. The irradiance, in solar ponds is transferred through the water, captured by the pond's dark-coloured bottom where the concentration of the salt water is the maximum. With an ambient temperature of about 293 K, the solar pond provides a low-temperature in the range of 343 - 353 K; which might be extracted by either pumping the hot brine at the bottom layer of the pond or through a heat exchanger system (Chen, 2010).



**Figure 2.1:** The Pyramid Hill solar pond in Australia (Chen, 2010)

### **2.2.2 Geothermal**

Geothermal energy (or heat from the Earth) is a renewable energy source, located in the Earth's crust. It can be found the world over, though the high-temperature energy required for energizing power-generating plants is found in relatively rare locations (DiPippo, 2012). One of its merits when compared with other renewable sources, like solar energy, wind energy and biomass; is the great potential in supply that is constant and not dependant on weather conditions. This makes power-generating stations using energy from geothermal sources very reliable and feasible for usage in base-load supply (Geothermal energy association, 2012; Oroz, 2013).



**Figure 2.2:** The world map of installed geothermal power plants capacity (Bertani, 2012)

There are enormous quantities of geothermal heat in existence, either in the form of hydrothermal (or produced) water situated in ‘geothermal aquifers and hot dry rocks’, which are artificially produced or upgraded underground reservoirs (Oroz, 2013). The ‘enhanced geothermal system’ can use the heat energy of the hot dry rock.

There has been a steady growth in installed power plants using thermal energy from geothermal sources and this growth is estimated to continue in the geothermal energy market worldwide at about 12% annually (Norden, 2011). Bertani (2012) reported that power generation from geothermal has increased from about 8.9 GW installed capacity in 2005, generating 55,709 GWh/ year to

about 10.9 GW in 2010, generating 67,246 GWh/ year and a projected 19.8 GW installed capacity by 2015. Figure 2.2 depicts the map of installed geothermal power plants capacity by country as at 2010.

### **2.2.3 Biomass**

The term 'biomass' refers to all the organic matters that possess stored energy through the process of photosynthesis (Alternative energy, 2014); which are 'combustible in nature' (Ashish and Mohapatra, 2013). It mostly exists in one form as plants and might be found in animal wastes through food chain; all of which can be used as energy resources through processes like combustion, pyrolysis and gasification among others (Alternative energy, 2014).

Numerous biomass fuels have been exploited in modern times including the wood products, by-products and crop residues such as cobs, husks, straws, etc., and animal refuse among others (McKendry, 2002; Ashish and Mohapatra, 2013).

Although solar, wind and mini/ micro hydro dominate the renewable energy forecasts; biomass fuels look promising due to new emerging technologies. They have been playing a vital role in the renewable energy sector despite the niche markets for solar, wind and mini/ micro hydro (Ashish and Mohapatra, 2013).

In recent times, biomass is becoming one of the widely utilized renewable energy sources, second only to hydropower in electricity generation (Alternative energy, 2014). It is such a broadly exploited energy source, perhaps as a result of its cost effectiveness and indigenous nature, which makes it account for

nearly 15% of the total energy supply in the world and as much as 35% in developing countries, and is customarily used for cooking and heating (Alternative energy, 2014). Broadly speaking, biomass-based technologies are becoming popular as they have an edge over other renewables. While the power generating process is the same, be it using biomass fuel or fossil fuel, it is the required equipment that differs.

#### **2.2.4 Industrial Waste Heat**

Industrial waste heat refers to thermal energy generated during industrial processes, which is unexploited. The major industrial waste heat sources include hot combustion gases; heated products exiting industrial processes and heat transfer from heat carriers among others (U.S. Department of Energy, 2008). The amount of industrial waste heat is poorly estimated, however numerous studies have quantified that about 20 - 50% of the industrial primary energy consumed is exhausted as waste heat (Johnson et al., 2008; U.S. Department of Energy, 2008; Lian et al., 2011; Fang et al., 2013). While during certain industrial processes, heat losses are unavoidable; improving the efficiency or integrating waste heat recovery-to-power technologies can diminish these losses. By industrial waste heat recovery-to-power applications, energy utilization and efficiency is improved, and significantly primary energy demand and greenhouse gases emissions are reduced (Patil et al., 2009; Fang et al., 2013).

The common uses for recovered industrial waste heat include using the heat internally to increase the efficiency of the industrial processes, for example



combustion air preheating (Zhang et al., 2001), furnace loads preheating (Hao et al., 2008) etc. or externally across various industrial or built-up uses, for example waste heat recovery-to-power generation (Carcasci et al., 2014; Nguyen et al., 2014), absorption cooling and space heating (Garimella et al., 2011; Garimella, 2012; Wu et al., 2014) among others.

## **2.3 Vapour Power Cycle**

Power generation and refrigeration are the two most important areas of the application of thermodynamics, both of which are typically accomplished using systems that operate on a thermodynamic cycle (Cengel and Boles, 2006).

Thermodynamic cycles can generally be classified into power cycles and refrigeration cycles. The device or system employed to generate net work output is frequently known as 'engine' and the thermodynamic cycle it operates on is known as 'power cycle' (Cengel and Boles, 2006). While the device or system employed to generate 'refrigeration effect' are known as 'refrigerators, air conditioners or heat pumps' and the cycle they operate on is known as 'refrigeration cycle' (Cengel and Boles, 2006).

Thermodynamic cycles can further be classified as either gas or vapour cycles, subject to working fluid phase. In the gas cycle, the working fluid stays in the gaseous phase through the whole cycle; while in the vapour cycle, the working fluid occurs in the vapour and liquid phases at different part of the cycle (Cengel and Boles, 2006).

## 2.4 Heat Engines

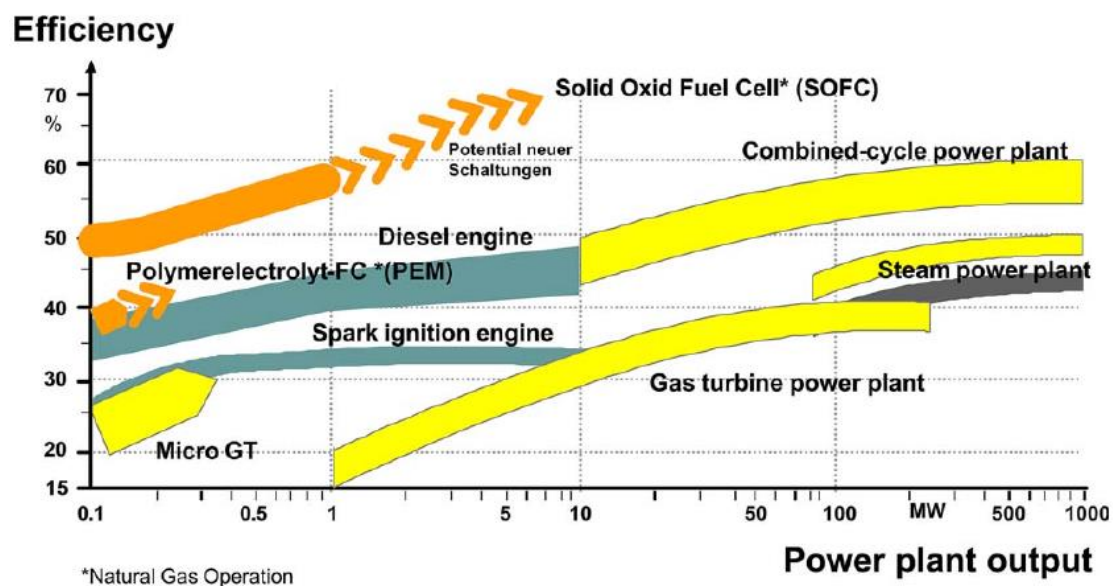
Heat engines differ significantly, but can all be characterized as follow; receive heat from a high-temperature source, convert part of this heat to work (usually in the form of a rotating shaft), reject the remaining waste heat to a low-temperature sink, and operate on a thermodynamic cycle (Cengel and Boles, 2006). For low-grade heat recovery-to-power employing an indirect (heat engine) conversion method, the Rankine-cycle engines using organic working fluids known as the organic Rankine cycles (ORCs), supercritical Rankine cycles (SRCs) and trilateral flash cycles (TFCs) are the major cycles that have been proposed and/or developed (Smith, 1993; Poullikkas, 2005; Fischer, 2011). They are classified as internal and external combustion engines; depending on how the heat is provided to the working fluid. In external combustion engines, like the steam power plants, heat is externally supplied to the working fluid. Heat engines are designed for the purpose of converting thermal energy into work, and their performance is expressed in terms of the 'thermal efficiency', which is the 'ratio of the net work produced by the engine to the total heat input' into the system (Cengel and Boles, 2006).

One of the key driving forces for innovative development in power generation is the efficiency improvement as well as the capacity gain, as illustrated in [Figure 2.3](#) for the different power-generating plants (Rukes and Taud, 2004). The steam and the combined-cycle power plants, at the upper end of the spectrum have attained their impressive unit limits. This implies that additional techno-economic development of these plants would typically be on efficiency improvement. The gas turbine power plants as well, with capacity limits of 250 -

300 MW, have attained impressive unit capacities. The micro-gas turbines or combined micro-gas turbines with heat recovery-to-power engines have an excellent market potential, provided it can be operated in a cogeneration mode.

Heat recovery-to-power technology presents an excellent capacity and efficiency improvement potential, where waste heat is used as energy source for sustainable power generation (Van Der Linden and Romero, 2009).

Overwhelmingly, power generation is mainly generated through the conversion of heat energy into electrical power using predominantly what is commonly known as a 'heat engine', which has the maximum thermal efficiency being constrained by the Carnot cycle (Brown et al., 2014).



**Figure 2.3:** Typical system efficiencies and output of power plants (Rukes and Taud, 2004)

### 2.4.1.1 Carnot Cycle

From fundamental thermodynamics, the ideal power cycle in relation to thermal efficiency is the Carnot cycle, which Sadi Carnot figured out in 1824 (Moran and Shapiro, 2004; Bejan, 2006). Carnot's ideal cycle yields the maximum thermal efficiency that any heat engine operating between a given hot reservoir (or heat source) at a temperature  $T_h$  and a cold reservoir (or heat sink) at a temperature  $T_c$ . However, it is essential to note that the Carnot cycle is required to function between a constant temperature heat and cold reservoirs; that is, a heat source and heat sink, whose temperatures are large enough such that the heat source could add or heat sink could receive any quantity of heat energy without any change in temperature (DiPippo, 2007). Moreover, Carnot's findings apply to a cycle that only comprises the reversible processes. Carnot's conclusions stem from the Carnot's theorem, which states that 'the efficiency of an ideal thermodynamic cycle can only depend on the temperatures of the heat source and heat sink, measured on the absolute temperature scale' (DiPippo, 2007).

The Carnot cycle comprises four reversible processes; which include the isentropic compression of working fluid during which work is performed on the cycle, constant pressure isothermal heat addition to the working fluid by a heating medium, isentropic expansion of working fluid during which work is produced by the cycle and constant pressure isothermal heat rejection from the working fluid to a cooling medium.

The thermal efficiency of a Carnot cycle operating between hot stream and cold stream reservoirs can be expressed as:

$$\eta_{th\_Carnot} = \frac{T_h - T_c}{T_h} = \left(1 - \frac{T_c}{T_h}\right) \quad (2-1)$$

Where  $T_c$  and  $T_h$  denote the temperatures of heat sink (or cold stream) and heat source (or hot stream) respectively.

It is an extremely idealized cycle, which requires an ideal, internally reversible heat engine although, furthermore, the heats transfer from the reservoirs is also externally reversible (Winterbone, 1997).

## 2.5 Heat Recovery-to-Power Cycles

The major techniques for heat recovery-to-power applications are direct and indirect conversion technologies (Bryson, 2007). Direct conversion technology, as its name implies, converts the heat energy directly into power without intermediate conversion and is thus free of any associated intermediate conversion losses e.g. gas lamp. While the indirect conversion technologies convert heat energy into power using heat engines. Although, the lack of extra conversion losses for the direct conversion is very attractive, the materials used and techniques employed result in lower efficiencies and power outputs compared with the main alternative, heat engine conversion (Bryson, 2007).

Low-temperature heat sources are defined in terms of their grade. These are classified into low, medium and high. There is no definite specification of the temperature range but can be classified within the temperature limit of 343 - 423 K for low, 423 - 773 K for medium (Peterson et al., 2008) and beyond 773 K is high (Harada, 2010).

Few decades ago, only high-temperature heat sources were useable for heat recovery-to-power generation in conventional steam power plants. At present, low-to-medium temperature heat sources are feasible for power generation using binary heat recovery-to-power technologies (Kalina et al., 1995; Boghossian, 2011).

Binary power plants use working fluids, which are usually organic fluids. The fluid works within a power cycle as follows: heat is transferred from the heat source to the pressurized fluid and the resulting vapour is expanded to derive shaft work (Boghossian, 2011). Various heat recovery-to-power engines for low-grade heat recovery-to-power applications have been proposed/ developed for low-grade heat conversion into mechanical or electrical power. The major ones are organic Rankine cycle (ORC), supercritical Rankine cycle and trilateral flash cycle (TFC) (Chen, 2011d). The success of the developed ones like the ORC technology might be due to the standardized units or dimensions used during construction of components, which allows flexibility and variety of use of components (Panesar, 2012). However, due to their wide-ranging applications in refrigeration, the technological maturity of these key components equally contributed to the success of the developed heat recovery-to-power technologies (Schuster et al., 2009). Therefore, the determining factors in heat recovery-to-power generation are their extent of availability, exergy of the temperature (or temperature profile) of the heat source, and the recovery and conversion cost of the waste heat-to-power (Panesar, 2012) among others.

### **2.5.1 Organic Rankine Cycle**

The steam Rankine cycle is a matured technology that Rankine in the 19th century had figured out its fundamental principles (Fischer, 2011). It is a closed vapour power cycle, which comprises the feed pump, evaporator, turbine, condenser and superheater where superheating is required. The Rankine cycle utilizes the gain of an insignificant quantity of work input needed to pressurise a working fluid and the amount of latent heat that can be recovered from a heat source (Harada, 2010) to power the cycle.

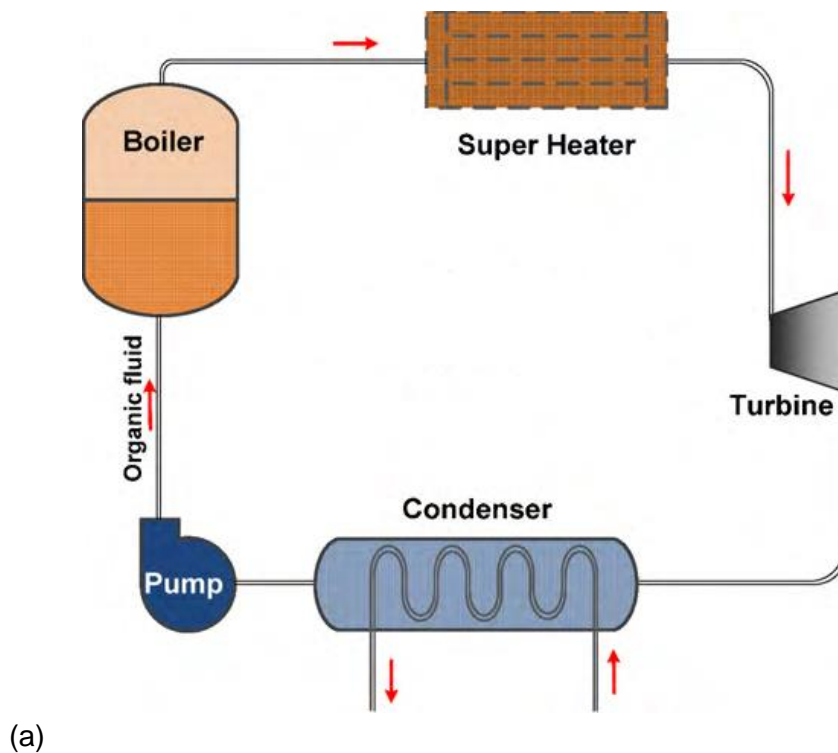
Organic Rankine cycle (ORC) differs from the steam Rankine cycle in that the working fluid is organic. ORCs are heat recovery-to-power cycles that apply the principle of the steam Rankine cycle, but employ the heated vapour of organic working fluid, instead of superheated steam (Sami, 2010; Sami, 2011), for heat recovery-to-power generation from low-grade heat source(s).

The applications of the ORC system comprise the solar thermal, geothermal, biomass, ocean thermal energy, waste heat recovery and desalination (Tchanché et al., 2011; Imran et al., 2014). The cycle has an average thermal efficiency limit of 2 - 11% and it is even lower with small-scale system lower than 5 kW (Kang, 2012; Li et al., 2013; Imran et al., 2014). Its thermal efficiency depends on system configuration and efficiency of the system components, working fluid, and the operating conditions of the cycle, heat source and heat sink (Imran et al., 2014).

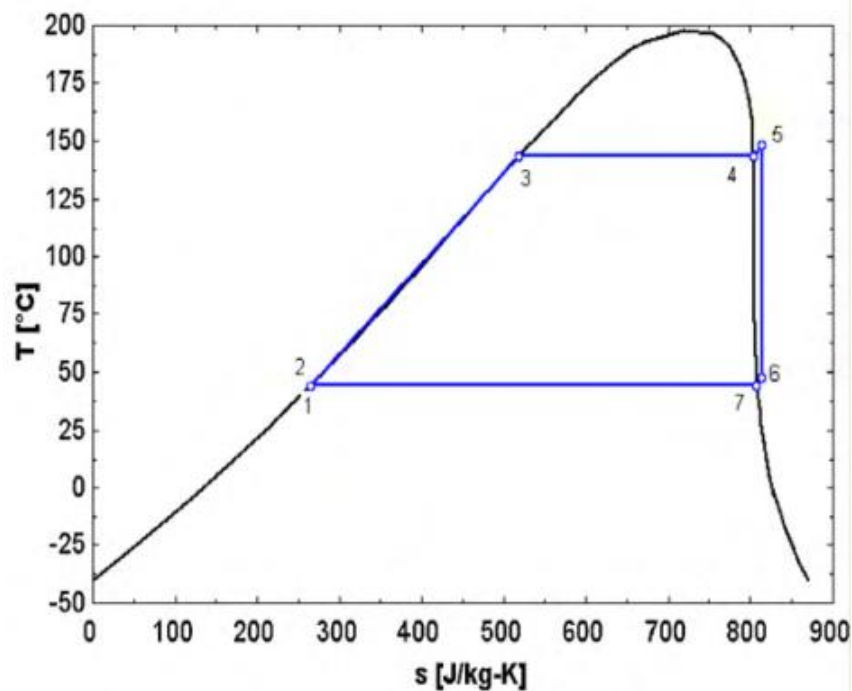
The ORCs have been largely examined and utilized to generate or co-generate power from low-to-medium temperature heat sources for their efficiency,

simplicity in the cycle configuration (Chen et al., 2011a; Liu et al., 2004; Bruno et al., 2008), dry expanding process and no emission of exhaust gas (Gu et al., 2009) among others. But a drawback of ORCs is the ‘isothermal boiling’ that causes poor thermal matching during the heat transfer from the heat source to the working fluid due to pinch-point (Chen et al., 2011a). The usage of zeotropic fluid mixtures (Maizza and Maizza, 1996; Maizza and Maizza, 2001; Wang and Wang, 2009; Borsukiewicz-Gozdur and Nowak, 2007) and supercritical fluids (Chen et al., 2005; Chen et al., 2006) can mitigate this challenge.

The cycle consists of the feed pump, boiler, turbine or expander, condenser and superheater (if superheating is required) (Chen, 2011b). Figure 2.4 depicts a schematic cycle configuration of an ORC and its thermodynamic process in a temperature–entropy ( $T$ – $s$ ) diagram.







(b)

**Figure 2.4:** The organic Rankine cycle, showing the cycle schematic configuration in (a) and the T–s diagram of its thermodynamic process in (b) (Chen et al., 2010)

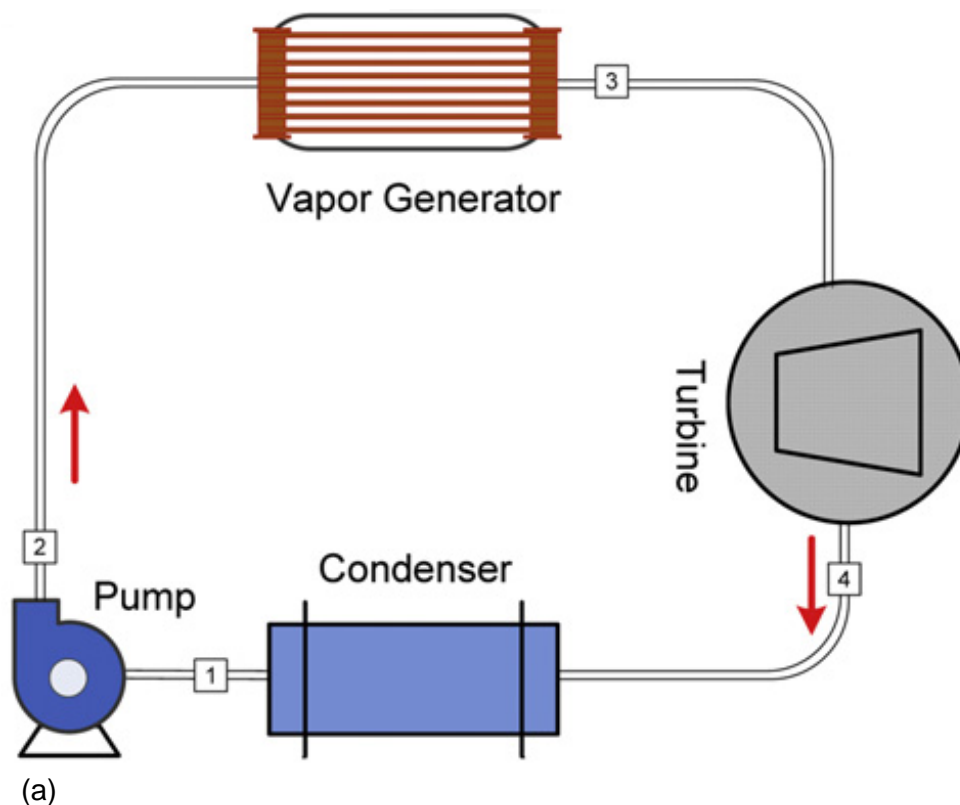
### 2.5.2 Supercritical Rankine Cycle

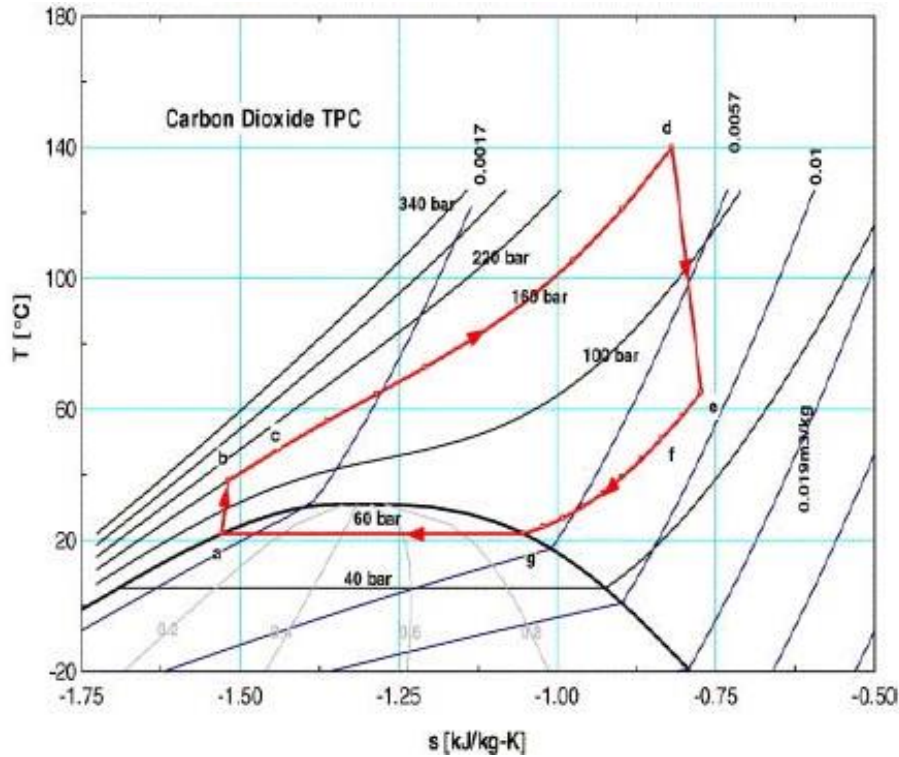
Supercritical Rankine cycles (SRCs) are heat recovery-to-power cycles that operate with a working fluid at its relatively low critical temperature and pressure, compressed to its supercritical pressure and heated directly from liquid state to its supercritical state before fluid expansion (Chen et al., 2011a).

Due to the bypass of the two-phase region, the SRC attains almost a perfect thermal match during the heat transfer from the heat source to the working fluid, minimizing exergy destruction (Karellas and Schuster, 2008; Chen et al., 2011a). The heating process does not undergo a distinctive two-phase region

like a conventional Rankine or ORC; consequently a better boiler thermal match is attained during the heat transfer from the heat source to the working fluid (Chen, 2011c).

The SRC has been examined to improve the performance of heat recovery-to-power from low-grade heat (Chen et al., 2006; Zhang et al., 2006; Karellas and Schuster, 2008). However, there is no SRC in operation, but it is becoming a new research direction due to its advantages in thermal efficiency and simplicity in configuration (Chen, 2011c). Figure 2.5 depicts the configuration of the CO<sub>2</sub> supercritical Rankine cycle and its thermodynamic processes in a T–s diagram.





(b)

**Figure 2.5:** The supercritical Rankine cycle, showing the cycle schematic configuration in (a) and the T–s diagram of its thermodynamic process in (b) (Chen et al., 2011a)

### 2.5.3 Trilateral Flash Cycle

Smith (1993) described a modified innovative heat engine (or thermodynamic) cycle for heat recovery-to-power application known as the trilateral flash cycle (TFC). The TFC has been studied in order to optimise heat recovery-to-power produced from a non-isothermal finite thermal energy source. This is based on the thermal matching and optimisation of the heat transfer from the heat source to the working fluid, cycle and load characteristics; heating the fluid to either its boiling point, to a saturated state or simply above its boiling point (Kelvin, 2008).

TFCs are heat recovery-to-power cycles in which the working fluid expansion starts from the saturated liquid condition rather than the saturated, superheated

or supercritical vapour phase. The TFC is a practical power cycle that mimics the Carnot cycle. It is a power cycle, which is especially suitable for heat recovery-to-power from non-isothermal low-to-moderate temperature heat sources (Brown and Mines, 1998). The cycle comprises four processes, which are the isentropic compression of working fluid during which work is performed on the cycle, constant pressure high-temperature heat addition to the fluid simply to its bubble point from a heating medium, isentropic expansion of the fluid that produces a two-phase mixture during which work is done by the cycle and constant pressure low-temperature heat rejection from the fluid to a cooling medium.

The key distinction between the TFC and the conventional binary cycles is that, the working fluid of the TFC remains in a liquid state as it leaves the heater, and the expansion of the fluid in the expander takes place completely as two-phase mixture in the two-phase region (Brown and Mines, 1998). The key to the practical implementation of the TFC is the efficient expansion of the pressurized liquid into the two-phase region.

Basically, it has the same components as the Rankine cycles but does not evaporate the working fluid; instead expands its saturated liquid, as a two-phase mixture by flash expansion in an expander for heat recovery-to-power generation (Zamfirescu and Dincer, 2008). Due to the heat transfer without pinch-point limitation, there is thermal marching between the exergy of the temperature profiles of the heat source and the working fluids (Winterbone, 1997; Zamfirescu and Dincer, 2008). Its major advantages are close thermal

match during the heat transfer from the heat source to the working fluid; it recovers more heat and achieves a higher working medium temperature than any other variant ORCs or flash steam systems (Smith et al., 2005).

Theoretically, the TFC has a lot of advantages in term of the efficiency and simplicity in configuration, which has made it become a subject of novel research. Moreover, this cycle is remarkable because it could perfectly match the exergy of the heat source temperature profile and function at rational pressures (Zamfirescu and Dincer, 2008). However, the difficulty of developing an efficient expander for two-phase expansion of the working fluid has been the main drawback. There is no TFC power system reportedly in operation (Chen, 2010) for obvious reason, because it is still at the research stage for technical development.

## **2.6 Working Fluid and its Selection Criteria**

Predictably, like the conventional Rankine cycles, the efficiency of a heat recovery-to-power system is restricted by the exergy destruction; i.e. its thermodynamic irreversibilities, as a result of entropy change within the system and environment during the different cycle processes (Hung, 2001). The irreversibilities and efficiency of the system depend on the working fluid and operating conditions; therefore the highest system efficiency can be attained when a promising (or suitable) working fluid is selected and operated at optimal conditions (Hung, 2001).

For any desired power output, the required size and achieved performance of the power cycle in performing shaft work depends on the working fluid's

properties (Badr et al., 1985). Thus, in any thermodynamic cycle, the working fluid performs a vital role in the processes of the cycle (Chen et al., 2010) and a careful selection that meets the cycle requirements is crucial for an 'efficient and safe operation' (Hung et al., 2010). It has its applicability range in line with its thermo-physical properties and a chemical stability in a desirable range of temperature. Moreover, operating conditions of any thermodynamic cycle, its system efficiency, economic feasibility and environmental impact are influenced by the fluid selection (Desai and Bandyopadhyay, 2009; Chen et al., 2010; Hung et al., 2010; Rayegan and Tao, 2011).

Organic working fluids normally possess a considerable high molecular mass that offers small volume streams, resulting in a system with compact size and that supports high expansion machine efficiency of over 80% (Sami, 2011). In contrast to expanded steam in the turbine, expanded organic fluids never form liquid droplets, preventing droplet impingement on the expansion machine blades and enabling flexibility in the heat exchanger design. One of the merits of using organic working fluid is that it simply needs a 'single-stage expansion machine', occasioning a 'simpler, more economical system in terms of capital costs and maintenance (Andersen and Bruno, 2005)'.

Furthermore, the key benefit of organic fluids as working fluid in heat recovery-to-power cycles, is that they can be powered at lower temperatures compared to the steam Rankine cycles and superheating is most often not required (Bruno et al., 2008).

The required criteria for selecting a working fluid for a power cycle are discussed to identify promising fluids for TFC engines. The key criteria and concerns for careful working fluid selection are as follow:

### **2.6.1 Thermo-Physical Properties**

The thermo-physical properties of the working fluid affect the efficiency of the power cycle and the power system's capital cost (Hung, 2001). The most significant thermo-physical properties necessary for a careful selection of a promising working fluid are as follow:

#### **2.6.1.1 Critical Points of Fluid**

The 'critical point of a working fluid is the peak point of the fluid saturation vapour line in a T-s diagram, which suggests the proper operating temperature and pressure limits for the working fluid in liquid and vapour forms' (Chen et al., 2010).

Literature is replete with numerous studies which suggest that recommended fluids are those with the highest critical temperature ( Hung, 2001; Maizza and Maizza, 2001; Mago et al., 2008; Dai et al., 2009b; Gu et al., 2009; Aljundi, 2011; Quoilin et al., 2011); that is, system performances as well as its efficiencies can further be increased by choice of the working fluids with higher critical point (Liu et al., 2004) particularly for a power cycle operating at the subcritical operating conditions.

In a power cycle, the process of condensation is vital and the condensation design temperature is usually over 300 K to permit low-temperature heat

rejection (heat sink) from the working fluid to a cooling medium or to the environment. Thus, fluids such as methane with a critical temperature lower than 300 K are not considered due to the difficulty posed by the condensing temperature of the heat sink (Chen et al., 2010). The operating pressure of fluid in a cycle must be within an acceptable range because very high pressure greatly affects cycle reliability and cost. The pressure values within the range 0.1 - 2.5 MPa and a roughly 3.5 pressure ratio are acceptable (Badr et al., 1985; Tchanche et al., 2009).

#### **2.6.1.2 Other Thermo-Physical Properties**

The flash and freezing points of fluid are also vital thermodynamic properties. The flash point of a working fluid should be high to prevent flammability while the freezing point should be lower than the lowest temperature of the operating cycle (Desai and Bandyopadhyay, 2009; Chen et al., 2010; Hung et al., 2010).

Fluids with high molecular weight, like dry or isentropic fluids, exhibit far lesser comparatively enthalpy drops during expansion compared with the water–steam mixture (Hung et al., 1997) due to their low specific enthalpy in vapour state (Smith, 1993). The higher the molecular weight, the smaller the expander required, consequently, fluids with high molecular weight are well-suited for system size reduction (Invernizzi et al., 2007).

#### **2.6.2 Fluid Stability and Compatibility with Materials**

Typically, at high pressure and high temperature, organic working fluids tend to chemically deteriorate and decompose (Quoilin, 2007), which could result in material corrosion and possible ignition (Hung et al., 2010). Thus, the chemical



stability of a fluid limits the maximum operating temperature of a cycle. Generally, working fluids are expected to be non-corrosive and thermally stable at operating pressure and temperature to avoid possible ignition and should be compatible with the component materials.

### **2.6.3 Environmental Concerns**

The ozone depletion potential (ODP) and global warming potential (GWP) are the two key environmental concerns, which symbolize the contributory potential of the working fluid to the degradation of the ozone layer and the warming of the globe respectively. Certain fluid usages have been terminated whereas others are being phased out by 2020 or 2030. Other working fluids, which have fascinating properties and do not pose any threat of environmental concerns like the phased out fluids, are being established. The major potential working fluids are obtained from fluids having fluorine and carbon atoms. The presence of one or more hydrogen atoms in the molecule, 'leads to it being mostly destroyed in the lower atmosphere by naturally occurring hydroxyl radicals, making sure that little or none of the fluid survives into the stratosphere' (Crook, 1994).

### **2.6.4 Availability and Cost**

One of the considerations for working fluid selection is its availability and cost. In ORCs, traditional refrigerants are employed as working fluids; which are quite costly. The utilization of cost effective hydrocarbons or a more mass manufactured refrigerants could make them available and reduce cost (Chen et al., 2010).

### 2.6.5 Types of Fluids

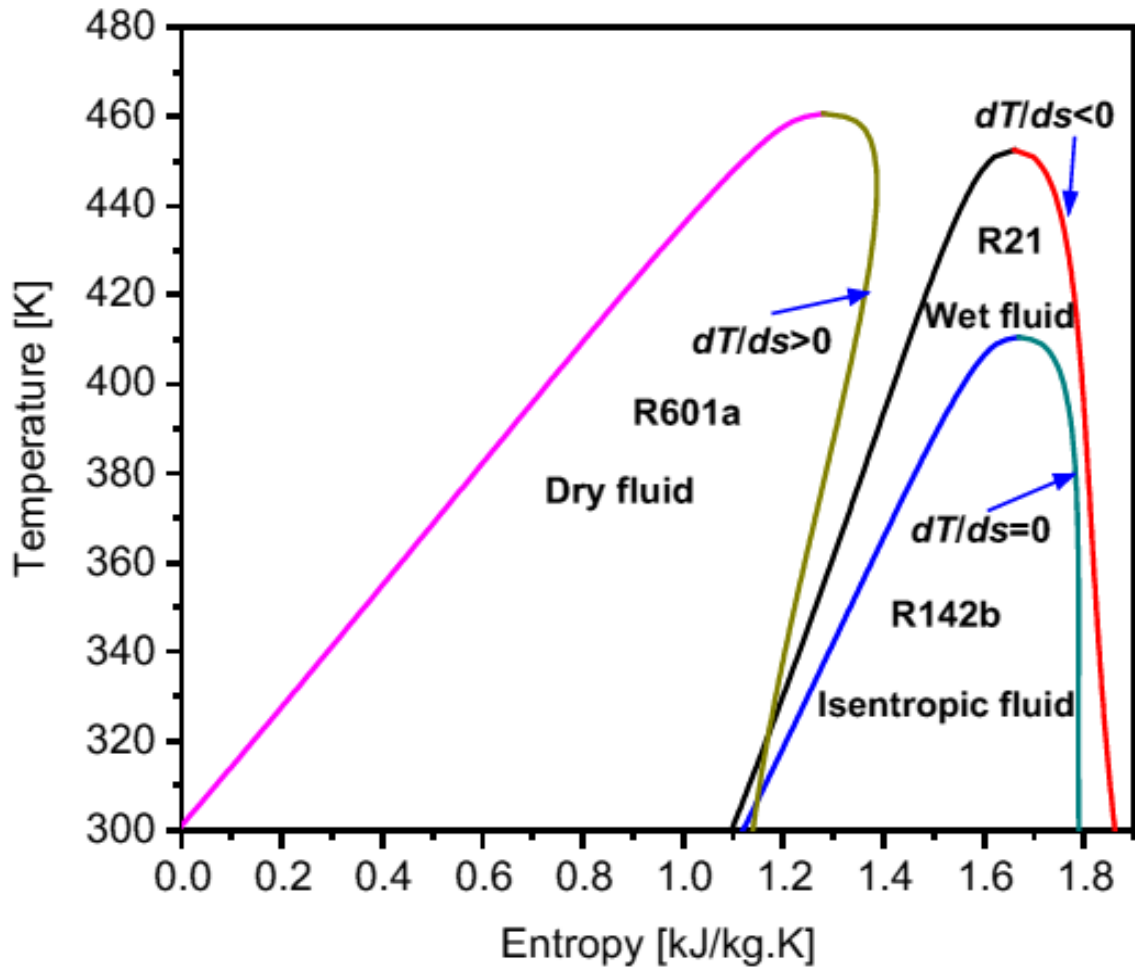
The vital characteristic of working fluids in heat recovery-to-power cycles is the vapour saturation curve, which influences the applicability of the fluid, efficiency of the cycle and related component arrangement or system configuration of the cycle (Hung et al., 1997). Working fluids are generally classified into three types depending on the shape of the slope of the vapour saturation curve ( $dT/ds$ ) on the  $T$ - $s$  diagram, which could be either positive, negative or vertical and the fluid is known as dry, wet or isentropic respectively (Figure 2.5) (Andersen and Bruno, 2005; Chen et al., 2010; Shengjun et al., 2011; Carcasci et al., 2014). As the value of  $dT/ds$  for the isentropic fluids tends to infinity, its inverse; that is,  $ds/dT$  is employed to depict the dryness or wetness a fluid (Chen et al., 2010).

Generally, the wet fluids e.g. water, ammonia, propyne etc. require superheating whereas dry fluids e.g. *n*-pentane, benzene, toluene, R-600, R-600a, etc. and isentropic fluids e.g. R-290, etc. that are mostly organic working fluids do not. This is because the isentropic expansion of saturated vapours of the dry fluids terminates at the superheated region while the saturated vapours of the isentropic fluids would retain their vapour saturated states throughout the expansion process without condensation (Hua et al., 1997). The property of persistent saturation through the expansion process and the fact that the need for a regenerator is overcome, establish the isentropic fluids as suitable working mediums in the TFCs as well as the ORCs and SRCs.

The use of wet working fluids in the TFCs as well as the ORCs is discouraged because they become saturated after a 'large enthalpy drop' in the expansion

machine and the fluid droplets impingement on the blades of the expansion machine during expansion posing a threat of damage to the turbine. Thus, isentropic or dry fluids are suggested to prevent the aforementioned problem (Shengjun et al., 2011). Though, the resulting expanded vapour leaves the expansion machine with a considerable quantity of 'superheat' if the working fluid is too dry, imposing a burden on the condenser. With this latent heat, if the working fluid is preheated after the feed pump pressurizes it, prior to its entry into the boiler; it could increase the cycle efficiency (Chen et al., 2010). This has been achieved through the integration of an internal heat exchanger or a regenerator in Rankine-cycle engines, which extracts the remaining latent heat at the exhaust of the expansion machine; reducing the heat exchanger and condenser loads, and as well improving performances.

The TFCs and SRCs still need much to study to obtain the potential working fluids. However, the use of wet fluids in SRCs is equally discouraged because it sub-cools and then nucleates to two-phase mixture, causing droplet impingement on the turbine blades, and lowering the turbine performance. With the dry or isentropic fluids, very fine droplets in the two-phase region are formed and no liquid poses a threat of damage to the turbine during expansion (Chen et al., 2010).



**Figure 2.6:** Working fluids types and their slopes of vapour saturation curve on the T–s diagram (Wang et al., 2013)

When the working fluid expansion process bypasses the two-phase region, fluid might leave the expansion machine with considerable quantity of ‘superheat’ if it is a dry fluid, imposing a burden on the condenser or necessitating a heat recovery scheme. If it is a supercritical fluid, system efficiency is maximized (Chen, 2011c) due to reduced desuperheating or because there is no need for it. For wet fluids, a ‘higher expansion machine inlet temperature’ would be required to prevent the expansion from undergoing the two-phase region with less concern over superheat during the process of condensation. However, with

the use of an expansion machine that involves two-phase region, a resulting potential net fluid effectiveness improvement in the range of 8% can be attained. Thus, in supercritical states of the SRCs, dry or isentropic fluids are preferred to wet fluids if the equilibrium states of the expansion process involve a two-phase region (Chen et al., 2010; Chen et al., 2011a).

In the saturated states of the TFC, light hydrocarbons are the most promising working fluids (Smith et al., 1995; Chen, 2011e).

### **2.6.6 Classification of Fluids**

Working fluids used in power cycles can be classified into inorganic fluid (e.g. water, ammonia, ammonia/ water, etc.) and organic fluid, which can further be categorised into hydrocarbons, refrigerants (non-hydrocarbon) and siloxanes (Bruno et al., 2008).

Inorganic fluids are one of the most commonly used fluids in heat recovery-to-power engines for power generation. But with the low-to-medium temperature heat sources, their performance as working fluids is either typically inefficient or not viable (Franco and Casarosa, 2002; Calise et al., 2014). This is specifically because of the negative slope of their saturation vapour curves on the T–s diagram (Calise et al., 2014). The organic fluids are characterized by a meaningfully better performance than the inorganic fluids powered by low-to-medium temperature heat source. This is due to their higher molecular weight, lower critical and ebullition temperatures (Quoilin et al., 2010), and positive slope of their saturation vapour curve on the T–s diagram (Calise et al., 2014).

The hydrocarbons fluids have had a wide-ranging application, from chemical applications to heat recovery-to-power generation and refrigeration. They display a strong positive slope of vapour saturation curve on the T–s diagram and a relatively high heat of vaporization. Although, these fluid are inflammable, which makes there uses in high-temperature environment challenging (Ilacqua, 2009). However, hydrocarbon fluids have been broadly utilized in comparative heat engines with no key challenges.

The refrigerants, on the other hand, have been widely used as a working fluid in refrigeration systems. These fluids possess critical temperature below 473 K and their shape of the slope of the vapour saturation curve on the T–s diagram could be either negative, which are wet fluids; or vertical, which are isentropic fluids. In addition, refrigerants have relatively low heat of vaporization, and as regards safety and environmental concerns, they have a high GWP, non-flammable and categorised as the safest among working fluid classes (Quoilin, 2007).

Siloxanes are organosilicon compounds, which originate from a combination of silicon, oxygen and alkanes. They comprise the  $R_2SiO$  units, where R denotes the hydrocarbon group or the hydrogen atom (Ilacqua, 2009). Invernizzi et al. (2007) examined siloxanes as working fluids in Rankine cycle and their performance metrics obtained depict comparative outcomes. Siloxanes display a strong positive slope of vapour saturation curve on the T–s diagram, are thermally stable at temperature limit as much as 673 - 723 K and have a relatively high molar complexity than the hydrocarbons.

## **2.7 Expander in Low-Grade Heat Recovery-to-Power Engine**

The low power-output expansion machines in heat recovery-to-power engines could be categorised into two main classes: turbines and positive-displacement machines (Badr et al., 1991a). Predominantly, positive-displacement machines such as volumetric expanders as expansion machines are one of the most important equipment in the heat recovery-to-power system, which are low-speed devices ( $< 5000$  rpm) applied in units of relatively low power output, in the range of 10 kW to 1 MW (Badr et al., 1984). They are possible alternatives to the single-stage turbines because the power output rises with increasing speed, whereas the reverse applies for turbines (Badr et al., 1991a; Smith et al., 2011).

Numerous numbers of devices have been examined for expansion of pressurized fluids into the two-phase region. There are five broad categories of these devices, comprising the Lysholm expanders, impulse expanders, total flow turbines, total flow systems and hero expanders (Brown and Mines, 1998). Lysholm expanders, also called helical or screw expanders, comprise the interlocking, counter rotating and helical rotors that rotate as the fluid is expanded along the axis of the rotors (Brown and Mines, 1998).

The expansion machines of different types for low power output have been evaluated and analysed based on their operational behaviours. Primarily, more attention is dedicated to the rotary machines among the positive-displacement category. This is because the reciprocating ones have numerous rotating components with associated inherent balancing problems; they display poor

breathing characteristics because of their high fluid-friction losses; have low efficiencies due to difficulties in lubrication particularly as in steam heat engines and are large and expensive to produce (Badr et al., 1991a; Badr et al., 1991b).

Experimental studies have shown that volumetric expanders are good candidates for low power output, due to their high efficiency, low rotating speed, reduced number of rotating components, reliability, wide range of power output, ready availability, simplicity in structure and good isentropic effectiveness (Guangbin et al., 2010; Quoilin et al., 2011). In particular, experimental studies have revealed very promising results on the scroll expanders modified from scroll compressors, with the reported isentropic effectiveness as much as 48 - 68% (Lemort et al., 2009; Quoilin et al., 2011). Another very promising candidate is the screw expander, which operates at a marginally higher output power showing the benefit of intake of a high liquid fraction at the inlet, giving room for the design of a wet cycle (Smith et al., 2011).

The expander is one of the main components of the heat recovery-to-power engines and its selection is a key element in heat recovery-to-power engine system design. The selection of the an appropriate expander depends on the working fluid thermo-physical properties, required mechanical power, mass and volumetric flow rates, and volumetric expansion ratio (Quoilin et al., 2010; Calise et al., 2014). The expanders are rotating machines that produce shaft (or mechanical) work by flash expansion of high pressure vapour (or saturated fluid) to a low condensation pressure vapour-liquid (Smith et al., 2011). Ideally,



because this expansion is isentropic, it not only decreases the pressure of the vapour but also its temperature.

### 2.7.1 Isentropic Efficiency

A complication is initiated when determining the specific enthalpy  $h$ , which is the 'sum of the internal energy  $u$ , and the product of pressure  $p$  and volume  $v$ ; i.e.,  $h = u + pv$  after a process if change of entropy is considered. Meanwhile, a process can be neither reversible nor adiabatic, so also there can be no change of entropy; which is always greater at the initial state of the process than at the end (Gentle et al., 2001).

Entropy can be simply expressed as that thermodynamic property, if it is plotted against absolute temperature, and the area under the process curve is the heat energy transferred (Gentle et al., 2001).

From this,

$$\delta s = \frac{\delta Q}{T} \quad (2-2)$$

Thus, the SI unit of the specific entropy is J/kgK or more typically, kJ/kgK.

From an engineering perspective, it is sufficient to say that as the change in entropy of a process tends to zero, the more effective it is. For example, more of the heat energy is converted into work after the expansion of gas, steam or vapour in a turbine or an expander, provided the change in entropy is minimal.

For any expander, isentropic efficiency indicates the change of entropy (Gentle et al., 2001). Its measured values for expanders differ generally and depend on the expander types, its expansion ratio and the thermo-physical property of the

working fluid (Brown and Mines, 1998). The ideal is 100%, denoting no change in entropy, but a usual isentropic efficiency limit is 75 - 90%.

## 2.8 Exergy: A Measure of Useful Work Potential

The concepts of energy and exergy stem from thermodynamics, which have found applications in all fields of science and technology. The introduction of a new energy quantity termed 'exergy' to work out in what manner work or power could be used considering a specified energy quantity with regard to the reference environment and/ or reasons for inefficiencies and the impossibility of ideal efficiency in all energy processes, are due to thermodynamic irreversibilities (Sato, 2004).

In engineering thermodynamics, exergy denoted by  $\Xi$  or B, is the maximum useful work done by a system for the duration of a transformation as the system comes to equilibrium with a reservoir or reference environment. Energy that is available to use is thus called exergy. The exergy of a system is equal to zero when the system and reference environment reach equilibrium (Pierre, 1998).

When a process involves a change in temperature, exergy is continuously destroyed. This destruction is proportional to the entropy rise of the system coupled with its reference environment. The exergy destroyed is known as **anergy**. Energy and exergy are interchangeable terms for a process or a change taking place at constant temperature (isothermal process) as exergy destruction does not arise (Exergy-Wikipedia Free Encyclopedia, 2011).

A Slovene mechanical engineer, Zoran Rant in 1953, coined the term exergy while Willard Gibbs developed the concept in 1873 (Wall, 1986). Exergy analysis was first conducted in the field of industrial ecology for a more efficient utilization of energy. Its application to unit operations in chemical plants in the 1900s, partly account for the enormous progress of the chemical industry. Then, it generally termed the available work or availability (Exergy-Wikipedia Free Encyclopedia, 2011).

### **2.8.1 Forms and Characteristics of Exergy**

The exergy and exergy efficiency of an assessed process or system for practical computational purposes can be classified into the following basic forms (Anđelković and Krstić, 2007):

**Kinetic exergy** can be expressed as the kinetic energy determined by virtue of velocity of motion with regard to the reference environment.

**Potential exergy** can be expressed as the potential energy determined by virtue of its position (zero level) with regard to the reference environment. Meanwhile, all the forces influencing the assessed matter and the reference environment are essential to determine the potential energy.

**Thermal exergy** can be expressed as the flow of exergy that passes through a substance of controlled volume, which is classified usually into two: physical and chemical exergy of flow.

The exergy of a substance is therefore conservatively categorised as the physical exergy, exergy related to temperature changes (thermal exergy),

pressure changes (pressure exergy, dynamic exergy), and concentration changes (mixing exergy), and chemical exergy, exergy related to changes in chemical composition of the substances (Sato, 2004).

### **2.8.2 Comparison of Energy and Exergy**

Energy is neither created nor destroyed but conserved during any processes, while exergy is dissipated in spontaneous processes. Energy depends upon the matter or energy flow parameters only, while exergy depends on both the matter or energy flow parameters and upon the parameters of the reference environment. Energy is based on the first law of thermodynamics, while exergy is based on the second law of thermodynamics coupled with mass and energy conservation principles. Lastly, energy is a measure of quantity, while exergy is a measure of quantity and quality (Odeyemi, 2010).

### **2.8.3 Exergy Analysis Method**

There are three different approaches to system analysis and performance evaluation based on the ‘first and second laws of thermodynamics’, which include the energy, entropy and exergy analyses (Sato, 2004). The energy analysis is absolutely a first law consideration, during which energy balances are performed to determine the energy flow rates into the system and out of it. The ‘analysis of entropy’ is also an exclusively second law consideration, which is associated with the mass flows and entropy generation as a result of energy transfer coupled with the processes transformation in a system. The exergy analysis or “exergy method of analysis” on the other hand however, is guided by a combination of the first and second laws of thermodynamics coupled with the

concept of irreversible production of entropy. For a proper assessment of a system performance, the incorporation of the second law consideration is necessary either as entropy or exergy analysis (Bejan, 1996).

Carnot and Clausius laid down the basics of exergy analysis in 1824 and 1865 respectively. The energy-related engineering systems are designed and their performance is evaluated primarily by using the energy balance deduced from the first law of thermodynamics. Engineers and scientists have traditionally applied the first law of thermodynamics to compute the enthalpy balances for quantifying the loss of efficiency in a process due to the loss of energy. However, in recent years the second law analysis, herein called the exergy analysis, of energy systems has more and more drawn the interest of energy engineers and the scientific community. The exergy concept has gained considerable interest in the thermodynamic analysis of thermal processes and plant systems since it has been seen that the first law analysis has been insufficient from an energy performance point stand (Bejan, 1996; Njoku et al., 2014). In this analysis, the heat does not have the same value as the work, and the exergy losses represent the real losses of work. When analysing novel and complex thermal systems, experience needs to be supplemented by more rigorous quantitative analytical tools. Exergy analysis provides those tools and helps to locate weak spots in a process. This analysis provides a quantitative measure of the quality of the energy in terms of its ability to perform work and leads to a more rational use of energy. In general, the specific exergy denoted by  $ex$  is calculated using the following equation (Ganapathy et al., 2009):

$$ex = ex_{ke} + ex_{p.e} + ex_{ph} + ex_{ch} \quad (2-3)$$

Where,  $ex_{ke}$  and  $ex_{p.e}$  are exergies due to motion or kinetic energy and exergy due to potential energy respectively,  $ex_{ph}$  is physical exergy i.e. exergy due to temperature difference and pressure difference with respect to the reference point and  $ex_{ch}$  is chemical exergy (i.e. due to chemical reactions).

Exergy analyses are thus effective for improving the energy efficiency of practically all manufacturing processes or systems. This is because it is (Ganapathy et al., 2009):

- i. An effective method employing the conservation of mass and conservation of the energy principles together with the second law for the design and analysis of energy systems;
- ii. An effective way to study how to make systems and processes more efficient;
- iii. An efficient technique revealing whether or not and by how much it is possible to design more efficient energy systems by reducing the inefficiencies;
- iv. A suitable technique for furthering the goal of more efficient energy-resource use;
- v. A key tool for determining the locations, causes, and magnitudes of wastes and losses;

- vi. A measure of usefulness, quality or potential of a stream to cause change;  
and
- vii. A tool for sustainable development.

## **2.9 Areas of Application of Energy and Exergy Analyses**

The development of exergy analysis began around the 1930s and has increased notably since the 1970s (Moran and Sciubba, 1994). A wide range of devices, processes and systems have been since examined, including many related to electricity generation, such as gas-turbine cycles (Tyagi et al., 2005), Kalina cycles (Borgert and Velasquez, 2004), Stirling cycles (Martaj et al., 2006), geothermal binary power plants (Dagdas, 2005), and cogeneration (Kwon et al., 2001) assessments have been conducted. Thermo-exergetic of the sugar mill processes (Duarte et al., 2004), exergo-economic of thermal, chemical, and metallurgical processes (Can et al., 2002; Seider et al., 2004), exergo-economic of manufacturing sector (Al-Ghandoor et al., 2010) and exergy-based environment (Salas et al., 2005) assessments have also been undertaken.

Numerous studies conducted by Al-Ghandoor et al. (2009), Rosen and Tang (2006a), Rosen and Tang (2006b), Rosen and Dincer (2003) and Horlock et al., 1998 have contributed to the principles, applications and practice of exergy analysis. Recently, Hepbasli (2008) carried out a key review on exergy analysis and assessment of renewable energy resources, while Rezac and Metghalchi (2004) outlined the history of exergy analysis. Moran and Sciubba (1994) provided a brief survey of exergy principles and analyses along with emphasis

on areas of application. They concluded that the exergy balance can be used to determine the location, type, and true magnitude of the waste of energy resources, and thus can play an important part in developing strategies for more effective fuel use.

## **2.10 Reversible Work and Irreversibilities**

Exergy is a valuable tool in assessing the 'quality of energy' and 'comparing the work potentials' of different energy sources or systems. Exergy based assessment alone is, no doubt, adequate for examining devices operating within fixed states because the final state is assumed always at dead state; which is apparently not true for engineering systems.

Reversible work and irreversibility (or exergy destruction) are two quantities that relate the apparent initial and final states of processes and serve as valuable tools in the thermodynamic analysis of a component or system (Cengel and Boles, 2006).

Reversible work  $W_{rev}$  is defined as 'the maximum amount of useful work that can be produced (or the minimum work that needs to be supplied) as a system undergoes a process between the specified initial and final states' (Cengel and Boles, 2006). This is the useful work output achieved (or input used) when the process within the initial and final states is performed in a totally reversible manner.



When the 'final state is the dead state, the reversible work equals exergy' (Cengel and Boles, 2006). For work consuming processes, 'reversible work represents the minimum amount of work necessary to carry out that process'.

The differential within the reversible work  $W_{rev}$  and the useful work  $W_u$  is owing to the irreversibilities in the course of the process, and this differential is known as irreversibility  $I$ . It expressed as (Cengel and Boles, 2006):

$$I = W_{rev,out} - W_{u,out} \quad (2-4)$$

Or

$$I = W_{u,in} - W_{rev,in} \quad (2-5)$$

In a reversible process, reversible and useful work terms are the same, so irreversibility is zero, which is expected because a reversible process does not generate entropy. While for a reversible process, the irreversibility is a positive quantity because  $W_{rev}$  is greater than or equal to  $W_u$  for a work producing device and  $W_{rev}$  is less than or equal to  $W_u$  for a work consuming device.

The wasted work potential or the lost work opportunity is regarded as irreversibility, which represents the energy that may perhaps be converted to work but was not. The smaller the irreversibility associated with a process, the greater the work that is produced (or the smaller the work that is consumed) (Cengel and Boles, 2006). The performance of a system can be improved by minimizing the irreversibility associated with it.

## **2.11 Review of Innovative Heat Recovery-to-Power Generation Systems**

Literature is replete with numerous studies conducted on the analyses and performance evaluation of energy conversion systems for power generation from conventional fuels, based on energy and exergy considerations. These have provided useful insights into the power cycles and, thereby, presented a basis for efficiency improvement efforts (Rosen and Tang, 2008), process integration and systemic analysis.

However, some extensive efforts have been devoted in the past two decades to the development and evaluation of the proposed innovative heat recovery-to-power engines, which are able to recover and convert into mechanical work, low-to-medium temperature heat sources such as hot exhaust gases from topping Brayton and/ or steam Rankine cycles, waste heat from industrial processes, geothermal sources, solar thermal, nuclear reactors or ocean thermal energy (Zamfirescu and Dincer, 2008) to mention a few.

Among these proposed heat recovery-to-power engines, there is a wealth of literature on the ORCs technology for heat recovery-to-power generation, which is typically categorized into the working fluids selection (Tchanche et al., 2009; Saleh et al., 2007; Lakew and Bolland, 2010; Bao and Zhao, 2013; He et al., 2014; Roy et al., 2010), modelling and simulation of the ORCs (Quoilin et al., 2011; Lemort et al., 2009; Wei et al., 2008; Lemort and Quoilin, 2009), applications of the ORCs (Tchanche et al., 2011; Hajabdollahi et al., 2013; Liu and Li, 2014), expanders modelling and design (Bao and Zhao, 2013; Lemort

and Quoilin, 2009; Giuffrida, 2014), and design and optimization of the ORCs (Roy and Misra, 2012; Quoilin et al., 2013; Imran et al., 2014). Considering the enormous number of published works on this cycle, the cited references are a far cry of what is available in the literature. While ORCs are used in existing power plants, the TFC is still at the research stage of technical development.

The following are the comprehensive review of the available published works in the literature on the TFCs technology for heat recovery-to-power generation. Smith, (1993) studied the basic principles for the development of the simple TFC for heat recovery-to-power. He found that, provided the two-phase expander can attain isentropic efficiency not less than 75%, the net work output of the cycle can be as much as 80% higher than simple Rankine-cycle engines for heat recovery-to-power from heat streams of temperature limit of 373 - 473 K. Moreover, the estimated cost per unit net work output is almost equal to that of Rankine-cycle engines. The preferred working fluids for TFC power plants are light hydrocarbons. Smith and da Silva (1994) investigated the proposed TFC engine using working fluid mixture as a means of increasing work output of the cycle for heat recovery-to-power from low-to-medium temperature heat sources. The results obtained depict that at the inlet temperature limit of 423 - 453 K and condensing temperature limit of 273 - 323 K, it is possible with a mixture of the *n*-pentane and neopentane for fluid to complete the expansion process as dry saturated vapour. The method of fluid property estimation was described and its accuracy confirmed by experiment. Smith et al., (1996) performed wide-ranging studies that has led to an improved understanding of how Lysholm twin-screw expansion machines might be used to derive

mechanical (shaft) work from two-phase flash expansion processes. The operating mode of the machine was described as well as the types of rotor shapes used. Process simulation results of the expansion process were analyzed and the effects of different working fluids, varying rotor profiles and sizes, and power outputs of 5 - 850 kW were investigated. There were good agreements between the predicted and measured performance metrics. Moreover, the statistical analyses of the results indicate that this is unlikely to be improved without the development of more refined methods of two-phase flow analysis. Brown and Mines (1998) carried out detailed computations to examine the applicability of a shortlist of 20 different working fluids in trilateral cycle (TLC) for heat recovery-to-power from geothermal brines at temperature limit of 366 - 433 K. The differentials in the net work outputs of almost 50% were computed for the different fluids at the lower temperature limits examined; while at the upper temperature limits, the differentials in the net work outputs were less pronounced. The working fluid selection the TLC is based on the temperature of heat source and any limitation on the fluid expansion ratio within the expander. The more volatile of the fluids evaluated had the lower expansion ratios and the computed results suggest that the TLC could use geothermal resource more efficiently than conventional binary cycles at the given temperature limit. In relation to a binary cycle, operating at an expander isentropic efficiency of 85%, the TLC has a comparative advantage if the two-phase expander isentropic efficiency about 76% or above. Bryson (2007) studied a modified heat engine using an innovative heat recovery-to-power cycle, known as TFC that utilizes an organic working fluid. The theoretical

results were compared with the empirical results and reasons for their discrepancies were investigated. Zamfirescu and Dincer (2008) assessed the thermodynamic performances of ammonia–water based trilateral cycle (TLC), which does not use a boiler, instead expands its saturated liquid by flash expansion in expander for heat recovery-to-power generation. The effects of the varying isentropic efficiency of the expander, concentration of the ammonia and the cooling fluid rate on system performances were examined. Fischer (2011) performed a comparison of the optimized TLC systems using water as working fluid and optimized ORC systems using pure organic working fluids. It was found that the exergy efficiency for heat recovery-to-power generation is higher by 14 - 29% for the TFC than for the ORC. Lai and Fischer (2012) presented model results for power flash cycles (PFCs) (a generalization of TLCs) engines including the heat addition to and rejection from different working fluids. A comparison of the TLC using water as working fluid, ORCs and water Clausius-Rankine cycles were made. General outcomes show that the PFC has higher efficiencies than ORC but larger volumetric flows at the exhaust of the expander. Also, the TLCs using water as working fluid have the highest heat recovery-to-power efficiencies. The drawback of using water is the large outlet volumetric flows at low-temperature limits. In these cases, light hydrocarbon like alkanes are promising working fluid in PFC due to their substantially smaller outlet volumetric flows and rather high heat recovery-to-power efficiencies. Wang et al., (2012) developed mathematical models for determining the performance metrics of a twin-screw expander in a simple TFC system using organic components. The geometry parameters of the twin-screw expander

associated with the angle of rotation for the male rotor comprising suction and discharge port area, leakage area and groove volume among others were employed in the modelling. The combined effects of internal leakage, oil injection, gas-oil heat transfer and thermo-physical property of the refrigerant were considered and their single parametric sensitivity studies were carried out. The models were validated with the analysed results of the diagram of the twin-screw expander experimental recording and there were good agreement between the theoretical calculation and the experimental data. This suggested that the models could be used for performance prediction and product development.

A critical review of the vital literature confirmed the need for development of the TFCs technology for low-grade waste heat recovery-to-power generation using unconventional working fluid. It underscored that, in recent years, though significant development has been achieved, largely in the working fluid selection, applications of TFC, and expanders modelling and design approaches particularly for twin-screw expanders. Research and development in the establishment of simulation and design tools of the TFC, its design and optimization, thermodynamic and parametric analyses for heat recovery-to-power generation, are required. This is because significant research prospects exist in modelling, simulation and evaluation of performance metrics of the TFC, its thermodynamic feasibility, system configuration design and optimization among others. These are needed for performance prediction and system development from the proof-of-concept towards a market-ready TFC technology for waste heat recovery-to-power generation.

Therefore, the need to establish simulation tools of the TFC for improving the understanding of its operation, physics of performance metrics and to evaluate novel configurations for low-grade waste heat recovery-to-power generation.

**CHAPTER THREE**

**MODELLING, SIMULATION AND EVALUATION OF  
TRILATERAL FLASH CYCLE SYSTEMS**



### 3 Introduction

Heat recovery-to-power has been a longstanding challenge that Carnot, Clausius and Rankine in the 19th century had figured out its fundamental principles (Fischer, 2011). Conventionally, mechanical power is recovered from external heat sources, such as combustion products, using Rankine-cycle engine that uses steam as the working fluid (Smith et al., 2011). It has gained prominence because of its very attractive features like good thermal integration with the topping cycle, highly reliable and extensive industry experience (Chacartegui et al., 2009). However, because the heat discharged are between 70 - 370° C (Hung et al., 1997), values not compatible with the conventional Rankine-cycle engine or due to the low conversion efficiency (Chen et al., 2011a; Liu et al., 2004); low-grade heat streams of many renewable thermal sources as well as the by-product (waste heat) of numerous combustion processes are ordinarily under-exploited.

One consequence of these, at present, is the intensified studies of heat recovery-to-power cycles using unconventional working fluid for low-grade heat recovery-to-power generation (Zamfirescu and Dincer, 2008; Zamfirescu and Dincer, 2009). This had resulted in a number of novel thermodynamic cycles being proposed for improved efficiency, reduced emissions and to gain on a smaller scale, a comparable advantage of efficiency, where performance of the conventional steam power plants are generally inefficient (Franco and Casarosa, 2002). These low-grade heat recovery-to-power cycles mainly include organic Rankine cycles (ORCs), supercritical Rankine cycles and trilateral flash cycles (TFCs).

Among these cycles, the ORCs have been broadly studied and employed to generate and co-generate power from low-to-moderate temperature (low-grade) heat sources for their efficiency, simplicity in the cycle configuration (Liu et al., 2004; Bruno et al., 2008; Husband and Beyene, 2008a; Husband and Beyene, 2008b; Wei et al., 2008; Chen et al., 2011a), ease of maintenance, improved part-load performance (Desai and Bandyopadhyay, 2009; Chen et al., 2010; Hung et al., 2010) and ability to be adapted to different heat source temperature profile (Delgado-Torres and García-Rodríguez, 2010). But a crucial limitation of the ORCs is the 'isothermal boiling' particularly with the pure fluids, which causes poor thermal matching during the heat transfer from the heat source to the working fluid due to pinch-point (Chen et al., 2011a). As a result of the pinch-point limitation, there is huge exergy destruction during the heat exchange process of the cycle.

The TFC is a heat recovery-to-power cycle at its basis proposed to enhance the performance of heat conversion into power particular from low-to-medium temperature non-isothermal heat source, i.e. of variable temperature. Basically, it has the same components as the Rankine-cycle engines but unlike the Rankine cycle, it does not evaporate the working fluid during the heating phase; instead expands it, from the saturated liquid condition, as a two-phase mixture. Due to the bypass of the isothermal boiling phase, there is a better thermal match during the heat transfer from the heat source to the working fluid (Zamfirescu and Dincer, 2008; Chan et al., 2013), which minimizes exergy destruction. Moreover, there is a growing interest in the cycle because it thermally matches the exergy temperature profiles of non-isothermal heat

sources and functions at moderate pressures such that its application is viable economically for shaft work or power generation (Zamfirescu and Dincer, 2008).

The choice of suitable working fluid for a given application, the feasible operating conditions/ parameters and the system configuration design, is one of the utmost essential factors that influence the performance of heat recovery-to-power cycles (Chen et al., 2011a; Sahin et al., 2013). The applicability range of the working fluid must be within its thermo-physical properties and its chemical stability in a desirable range of temperature (Chen et al., 2010). The cycle high temperature upper limit depends on the fluid thermal stability and its lower limit on the techno-economic factors (Boghossian, 2011) such as the size of the heat exchangers required for the heat addition to and rejection from the working fluid.

With the application of suitable working fluid, the TFC system can be designed to recover and convert low-grade heat efficiently from the renewable thermal sources as well as from the by-product (waste heat) of numerous non-renewable sources into electrical power. Its power generating capacity for a unit could span from few kilowatts to few megawatts, while poly-units in parallel could form multi-megawatt thermal power plant.

### **3.1 Methodology**

A design criterion for the trilateral flash cycles (TFCs) power plants was established subject to the engineering design constraints. The system configuration designs of the TFCs for waste heat recovery-to-power generation were configured such that the cycles recover and convert waste heat from a

broader application, especially for the evaluation outside the boundaries of known application.

The TFCs, which consist of the simple TFC and proposed improvement schemes of the process integration of the TFC to optimize the cycle configuration design and/ or operation for improved heat exchange performance and system efficiency were studied.

The system configuration designs considered were the simple TFC, TFC with internal heat exchanger (IHE), TFC with reheating otherwise known as reheat TFC and TFC with feed fluid-heating also termed regenerative TFC.

A semi-empirical modelling approach that involves limited number of meaningful parameters, which were easily distinguished for the system configuration design and sizing, experimental measurements for prototype development and parameters estimations, performance predictions as well as for the evaluations of the different applications, was adopted. This is because such models are easily interconnected to build a larger process simulation model (Quoilin et al., 2010).

The governing equations and complete balance equations, i.e. mass, energy and exergy rate balances were formulated for each individual cycle components.

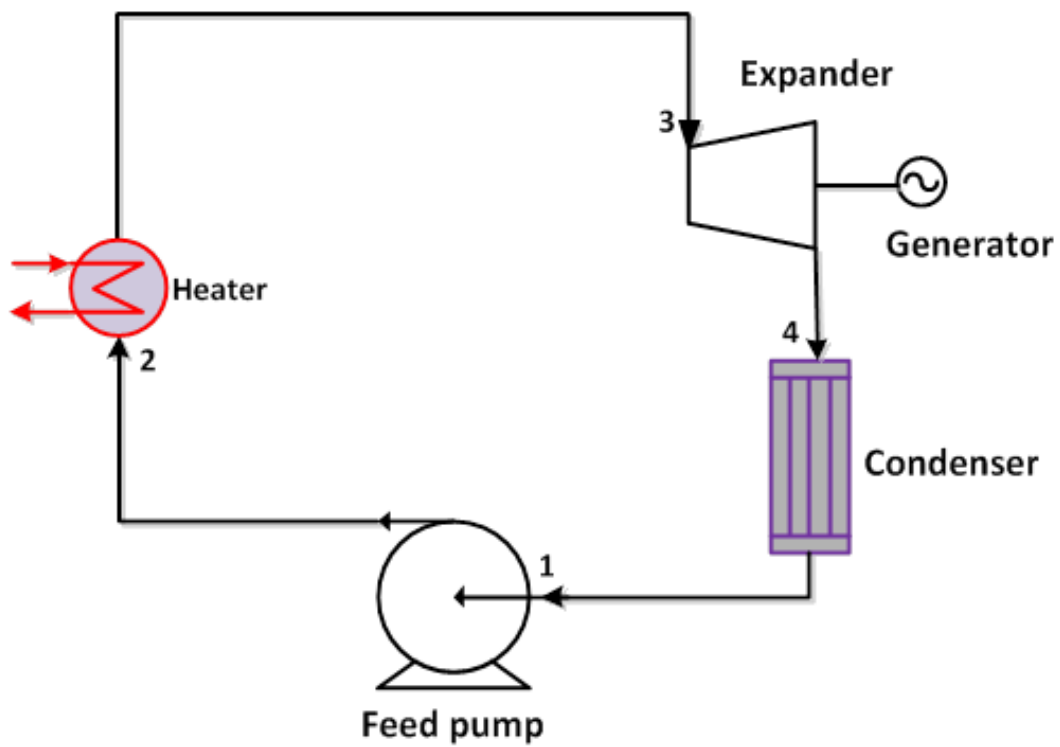
Steady-state steady-flow models of the TFCs power plants, corresponding to their thermodynamic processes were thermodynamically modelled and implemented employing EES (engineering equation solver) (Klein, 2013). The

process simulations of the models of the cycles and ancillary components at the subcritical operating conditions were implemented using the defined model input parameters and results obtained. The performances of the cycles were evaluated using an expander inlet pressure limit of 2 - 3 MPa and expander inlet temperature limit of 393 - 473 K at the average condensing temperature of 309 K. The results were analysed for all cycles and their design base were established within the viable operating (working) conditions.

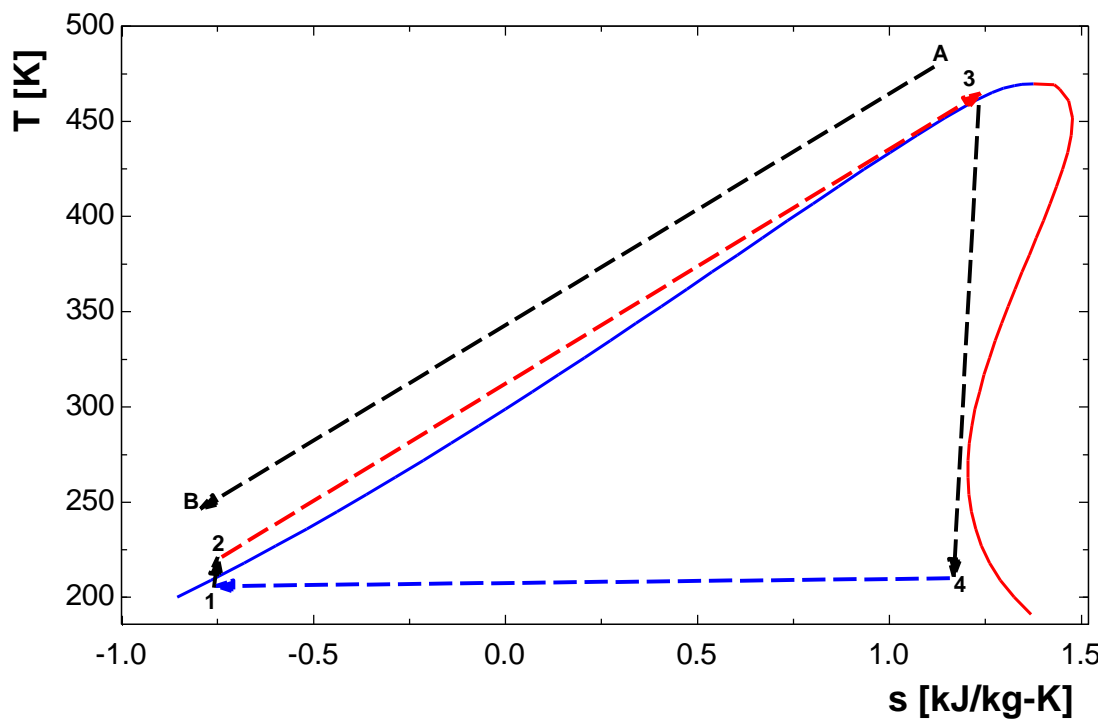
### **3.1.1 System Description**

Figure 3.1 depicts the simple TFC schematic cycle configuration and the temperature–entropy (T–s) diagram of its thermodynamic process. It comprises four key components, which are the feed pump, heater, expander and condenser.

Like the Rankine cycles, the working fluid of the TFC is a saturated liquid at the ambient (reference) temperature and pressure (state 1) supplied to the feed pump where it is pressurized (state 1 - 2) to a higher pressure. Afterward, heat is added to the fluid (state 2 - 3) just to its saturated temperature (boiling point) by a heating medium in the heat exchanger at constant pressure and is being injected into the expander; where shaft work is produced by flash expansion of the high pressure saturated fluid to a low condensing pressure (state 3 - 4), driving a generator to produce electrical power. Subsequently, the resulting vapour–liquid content is condensed to liquid (state 4 - 1) (by cooling agent that enters the condenser) to start the new cycle.



(a)



(b)

**Figure 3.1:** The simple trilateral flash cycle, showing the schematic cycle configuration in (a) and the  $T$ - $s$  diagram of its thermodynamic process in (b)

### **3.1.2 Improvement of the Trilateral Flash Cycle**

The improvement scheme (or modification) of the optimised design of the TFC considered in this study uses the concept of process integration of the simple TFC with an internal heat exchanger (IHE) (heat recuperator), reheating and regeneration (regenerator); which have been used in the conventional steam power plants. The following modified cycle configurations of the TFC integrate the aforesaid innovative efficiency improvement scheme.

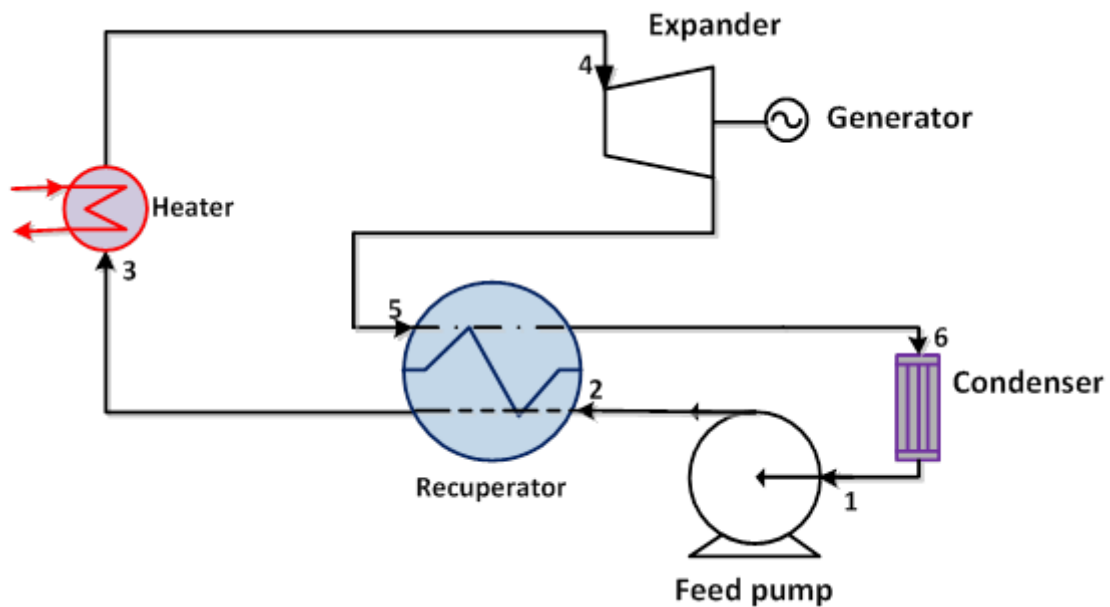
#### **3.1.2.1 Integration with Internal Heat Exchanger**

When the expander expansion process terminates in superheated region, the integration of an IHE at the exhaust of the expander might be beneficial for preheating the working fluid prior heating to the saturated temperature in the heat exchanger. The integration of an IHE, which extracts the remaining latent heat at the expander's exhaust would reduce the heat exchanger and condenser loads, and as well improve performances.

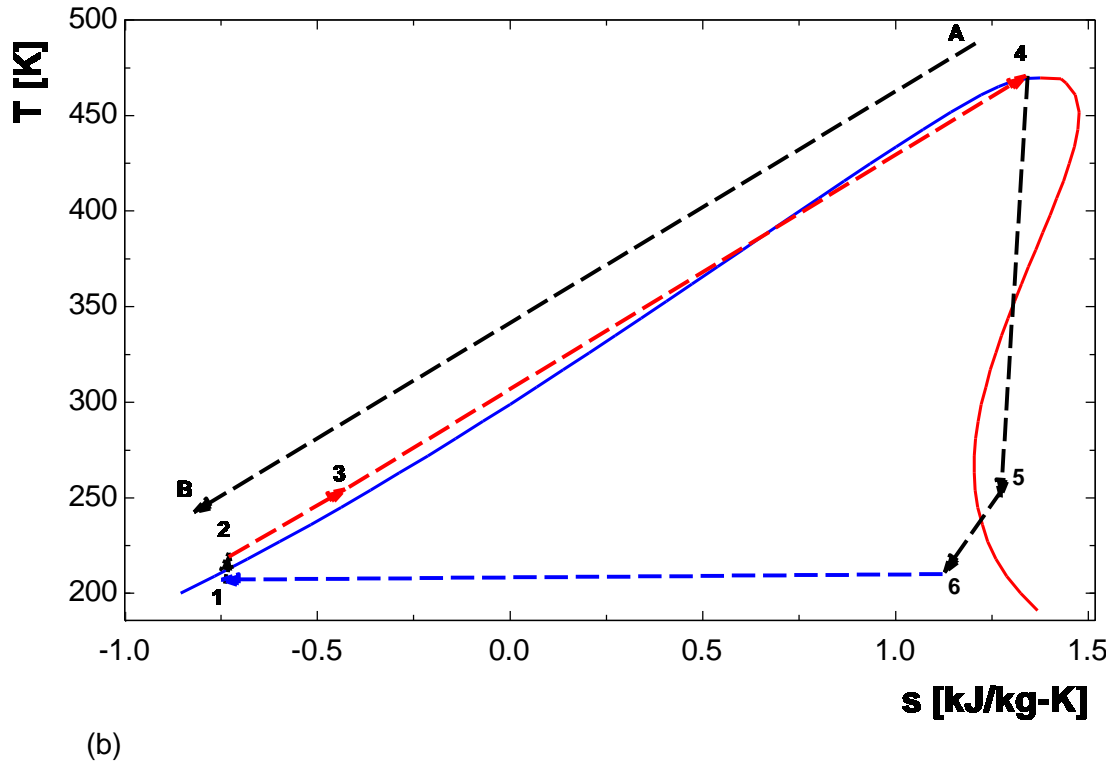
Thus, the integration of IHE to the simple TFC is considered for improvement of the heat exchange performance and system efficiency. The performance study of the effects of the integration of the IHE on simple TFC using organic working fluid is examined.

Figures 3.2 depict the schematic configuration of the TFC with IHE and the T–s diagram of its thermodynamic process. It comprises five key components, which are the feed pump, heater, expander, IHE and condenser. The latent heat extracted from the superheated vapour–liquid at the expander outlet by the IHE

is used to preheat the sub-cooled liquid at the feed pump outlet (state 2 - 3). Afterward, the fluid is heated to its saturated temperature (state 3 - 4) by a heating medium in the heat exchanger at constant pressure and is being injected into the expander (state 4 - 5), where shaft work is produced. Subsequently, the latent heat of the resulting vapour-liquid content is being bled (state 5) and condensed (state 6 - 1) by a cooling medium in the condenser to start the new cycle.







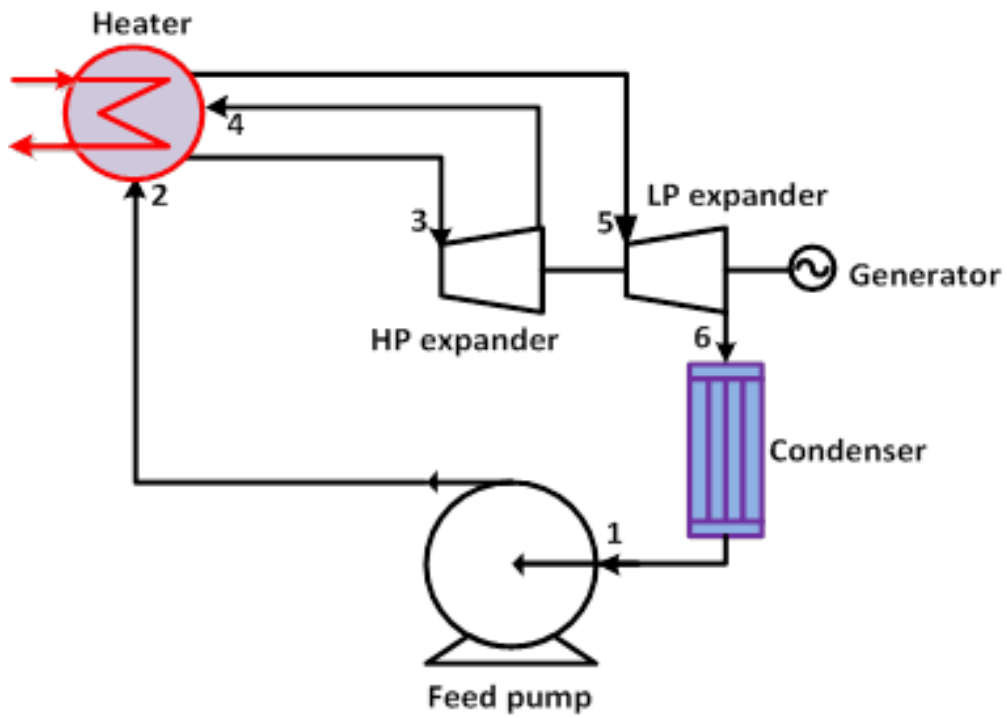
**Figure 3.2:** The trilateral flash cycle with IHE, showing the schematic cycle configuration in (a) and the T–s diagram of its thermodynamic process in (b)

### 3.1.2.2 Reheating

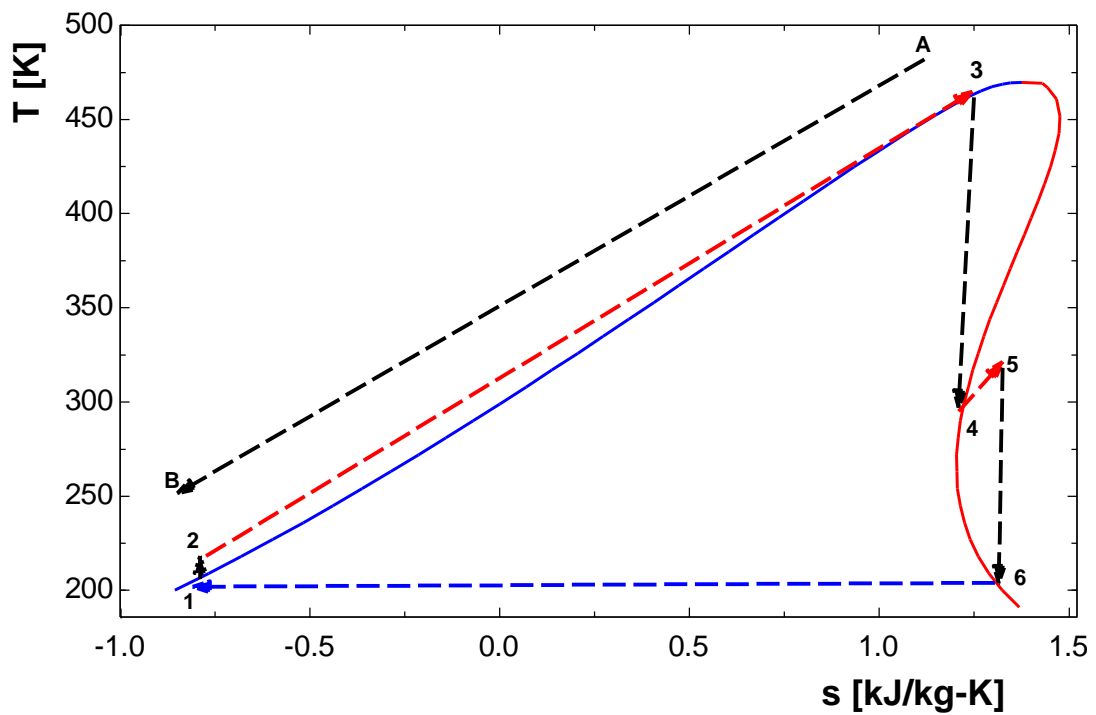
The reheating is a technique used to increase the expansion work of a thermodynamic cycle or process. The expansion process is completed in stages while the reheating of the working fluid is in between these stages. Since the steady-flow compression work is proportional to the specific volume of the flow, the specific volume of the working fluid should be as large as possible during an expansion process. Reheating is a practical solution to the excessive moisture problem in the lower-pressure stages of expander, and it is used frequently in modern vapour power cycles (Cengel and Boles, 2006).

Once the working fluid used in a TFC is a wet fluid for instance water, a reheating would be required to improve fluid dryness because it becomes saturated after an enthalpy drop in the expansion machine while the fluid droplets impingements on the blades of the expansion machine during expansion pose a threat of damage (Chen et al., 2010). Thus, to improve fluid dryness, the expansion process is split into two, and a reheating is introduced between both expansion stages. For performance study of the effects of reheating on TFC system, the TFC with reheating using organic fluid was examined.

The reheat TFC is a modification of the simple TFC, in which the working fluid is expanded in the expander in two stages and reheated in between. Figure 3.3 depicts the schematic cycle configuration of the reheat TFC and the T-s diagram of its thermodynamic process. It comprises five key components, which are feed pump, heater, high pressure (HP) expander, low pressure (LP) expander and condenser. Like the simple TFC, the high pressure saturated fluid is first expanded in the HP expander (state 3 - 4) and the medium pressure fluid is then returned to the heater where it is reheated to its saturated temperature (state 4 - 5) by a heating medium. Afterwards, the medium pressure saturated fluid is expanded in the LP expander (state 5 - 6); where shaft work is produced, with the resulting vapour-liquid content then condensed (state 6 - 1) by a cooling medium in the condenser to start the new cycle.



(a)



(b)

**Figure 3.3:** The trilateral flash cycle with reheating, showing the schematic cycle configuration in (a) and the T-s diagram of its thermodynamic process in (b)

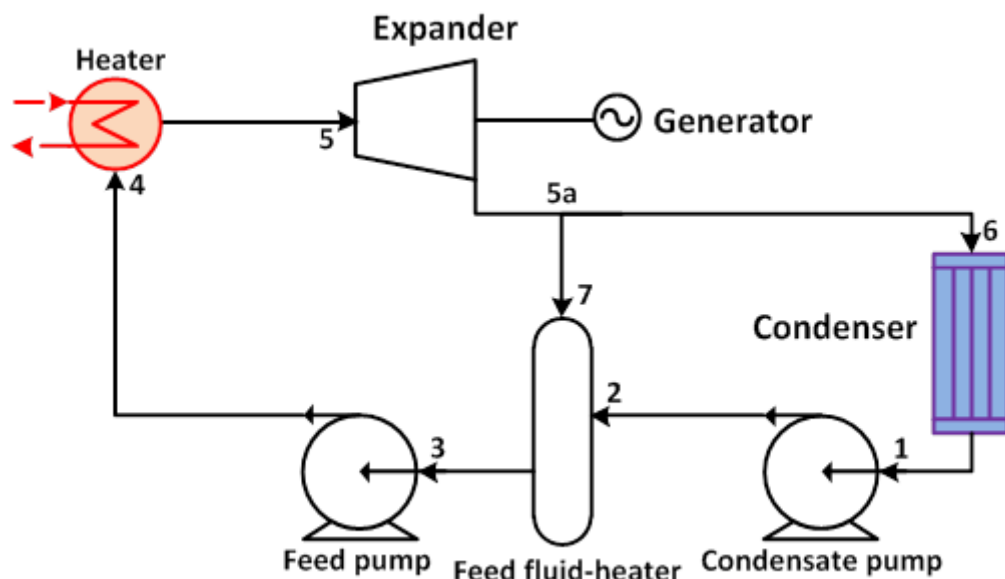
### 3.1.2.3 Regeneration

The regeneration is the process of transferring energy within a cycle from a working fluid at high temperature in part of the cycle to a lower temperature in another part of the cycle to reduce the amount of external heat transfer that is required to power the cycle. This technique is used to raise the temperature of the liquid leaving the pump (by the latent heat of the feed fluid) before it heated by the heating medium in the heat exchanger (Cengel and Boles, 2006). For instance, in steam power plants, 'extracting' or 'bleeding' steam at different points from the turbine is performed to attain a practical regeneration process. This steam, which could have produced more work by expanding further in the expansion machine, is used to preheat the feed water instead (Cengel and Boles, 2006).

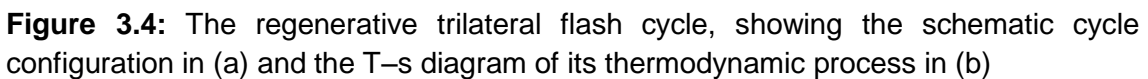
For improved heat exchange performance and system efficiency, the regenerative TFC using an open feed fluid-heating for preheating the working fluid was considered. Thus, the performance study of the effects of feed fluid-heating on TFC system using organic fluid was examined.

The regenerative TFC is a modification of the simple TFC, which is accomplished by 'extracting' or 'bleeding' vapour from the expander. Figure 3.4 depicts the schematic cycle configuration of the regenerative TFC and the T-s diagram of its thermodynamic process. It illustrates the system working principle with a bleed point for preheating the working fluid in an open feed fluid-heater. A fraction of the vapour flow rate is bled (point 5a) at the transitional pressure between the heating and the condensing pressure, which is directed to the heat

exchanger to preheat the pressurised feed fluid. The working fluid is pressurized with the condensate pump (state 1 - 2), then preheated in the feed fluid-heater (state 2 - 3) and is being pressurized to a high pressure with the feed pump (state 3 - 4). The high pressure fluid is heated (state 4 - 5) by a heating medium in the heat exchanger and injected into the expander (state 5 - 6); where it is expanded, with the resulting vapour-liquid content being bled (point 5a) for the preheating of the working fluid (at point 7) and condensed (state 6 - 1) by a cooling medium in the condenser to start the new cycle.



(a)



The thermodynamic processes of any cycle are illustrated with the pressure–enthalpy (p–h), pressure–volume (p–v) or T–s diagrams. The T–s diagram is typically used because it is valuable as a visual aid in the analysis of ideal power cycles: cycles that do not involve any internal thermodynamic irreversibilities.

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the heat exchanger (process 2 - 3), isentropic expansion of working fluid in the expander (3 - 4) during which work is done by the cycle and constant pressure heat rejection from the working fluid to a cooling medium in the condenser (process 4 - 1). States 3 and 5 in Figure 3.2 (b) depicted the integration of an IHE to the simple TFC.

Unlike the simple TFC, the reheat TFC comprises three constant pressure heat exchange processes, an isentropic compression process and two isentropic expansion processes (Figures 3.3). The reheating is depicted by states 4 and 5 on the T-s diagram of the cycle (Figure 3.3 (c)).

The regenerative cycle comprises three constant pressure heat exchange processes, two isentropic compression processes and an isentropic expansion process (Figure 3.4). Typically, the preheating with the expander bled and the condensate at the exit of the open feed-fluid heater is assumed at saturated condition. States 5a and 7 on the T-s diagram of the cycle (Figure 3.4 (d)) depict the regeneration process.

On a T-s diagram, the heat addition process proceeds in the direction of increasing entropy, while the heat rejection process proceeds in the direction of decreasing entropy, and the isentropic (internally reversible, adiabatic) process proceeds at constant entropy.

### **3.2 Working Fluid Selection**

The preliminary cycle evaluations and an in-depth investigation of the promising working fluids that can be used in the TFC were performed using the in-built thermodynamics and transport properties of working fluids in EES database.

Primarily, the working fluids screened include the 'linear, branched and cyclic hydrocarbons, refrigerants and siloxanes'. Of the initial shortlist of promising fluids, refrigerants containing chlorine were screened out due to prohibition with respect to their ozone depletion potential. Fluids that displayed a wet or a negative slope of vapour saturation curve on the T–s diagram were also screened out to ensure that the fluid after expansion tends to dry out.

Research have shown that for a heat source and heat sink profiles, light hydrocarbons are preferable working fluids for TFC systems based on the favourable considerations of the thermo-physical properties (relatively high specific enthalpy drops, low liquid density), thermal stability and environmental protection, system cost and expander design (Smith, 1993; Smith and da Silva, 1994). Subsequently, numerous categories of the light hydrocarbons comprising alcohols, aldehydes, alkanes, alkenes, alkynes, cyclic hydrocarbons, halocarbons, ketones and organometallic compounds were therefore considered in the search for a suitable organic working fluid for the TFC power plants. Of these fluids, a shortlist of 24 potential working fluids was evaluated based on their slope of vapour saturation curve ( $dT/ds$ ) on the T–s diagrams, thermo-physical properties, fluid stability and compatibility, environmental concerns, and availability and cost.

The vital characteristic of working fluids in heat recovery-to-power cycles is the vapour saturation curve, which influences the applicability of the fluid, efficiency of the cycle and the related component arrangement or system configuration of any cycle (Smith, 1993). Working fluids are generally classified into three types



depending on the shape of the  $dT/ds$  on the  $T-s$  diagram, which could be positive, negative or vertical and the fluid is known as dry, wet and isentropic respectively (Hung et al., 1997).

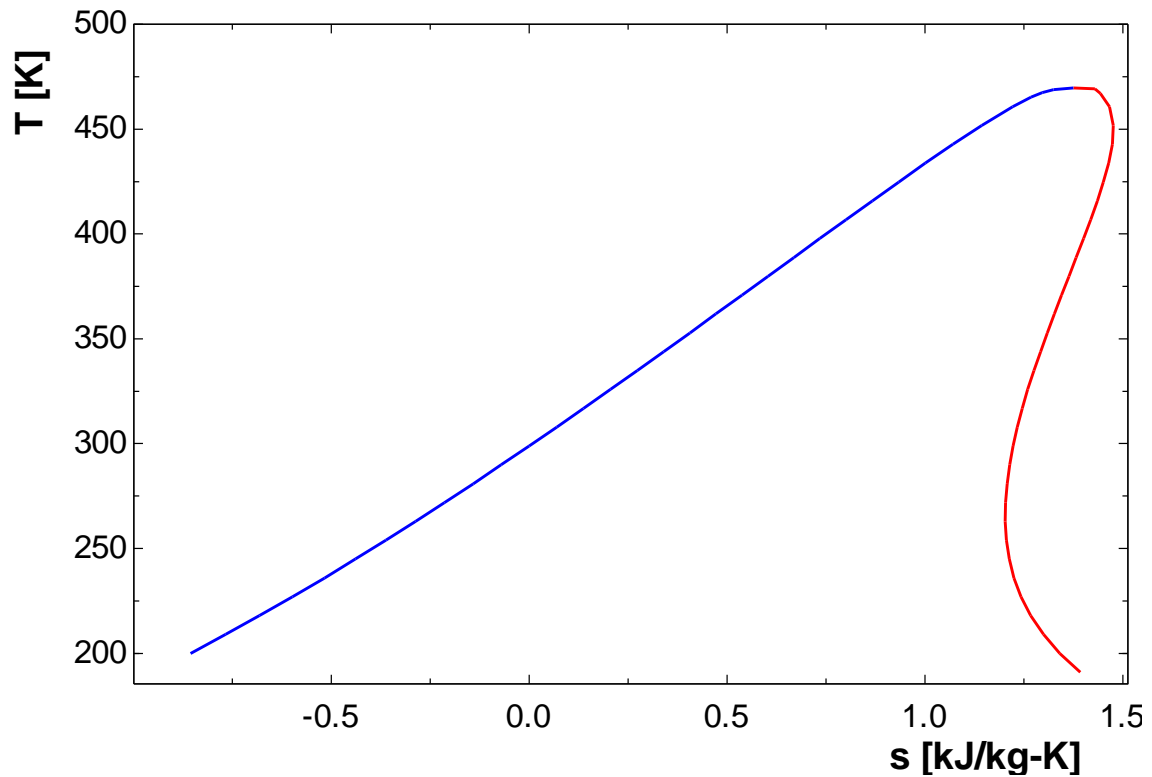
Not any of the alcohols, aldehydes, alkenes, alkynes and ketones had the requisite positive slope on the  $T-s$  diagram. The familiar halocarbons, because of their high molecular mass are all unacceptable, even though R-216 has about the right positive slope on the  $T-s$  diagram. This is because fluids with high molecular mass exhibit far lesser comparatively enthalpy drops during expansion (Chen et al., 2010; Andersen and Bruno, 2005; Shengjun et al., 2011) due to their low specific enthalpy in vapour state (Hung et al., 1997). Similarly, the outcomes of the cyclic hydrocarbons considered, comprising the cyclopropane, cyclobutane and cyclopentane are unacceptable. While the simplest of organic compounds, alkanes display a strong positive slope of the vapour saturation curve on the  $T-s$  diagram from pentane upward and the liquid pentane expansion tends to dry out at temperatures slightly exceeding 453 K (Smith, 1993). Thus, the focus of the search is on the isomers of butane and pentane, and further consideration was given to *n*-pentane, neopentane and isopentane.

The *n*-pentane was adopted as working fluid for this study because of its strong positive slope on the  $T-s$  diagram (Figure 3.5) and its saturated liquid expansion tends to dry out at temperatures slightly exceeding 453 K (Smith and da Silva, 1994). It possesses good thermo-physical properties, availability and low-cost, a boiling point slightly above the room temperature and relative safety

compared with the other isomers of pentane for low-to-medium temperature heat recovery-to-power applications. Moreover, *n*-pentane is a dry fluid (Desai and Bandyopadhyay, 2009) with low latent heat and specific volume, whose thermo-physical properties are well-suited for heat recovery-to-power generation from low-to-medium temperature heat sources. Table 3.1 listed some of the properties *n*-pentane.

**Table 3.1:** Properties of *n*-pentane (F-Chart Software, 2012)

Molecular mass [kg/kmol]	Critical temperature, $T_c$ [K]	Critical pressure, $P_c$ [kPa]	Boiling point temperature, $T_{bp}$ [K]
72.15	469.5	3364.0	309.0



**Figure 3.5:** Temperature–entropy diagram of *n*-pentane illustrating its positive slope of the vapour saturation curve (Klein, 2013)

### 3.3 Formulation of Governing Equations

The governing (general balance) equations for each component of the TFC systems were established depending on the heat, work and mass; energy and exergy considerations as well as the expressions for energy and exergy efficiencies for the TFC systems. The first-law (or energy) and second-law (or exergy) relations for the system components were employed to determine the energy terms, which include the heat addition and work output, energy and exergy efficiencies, and exergy load and irreversibilities.

Assuming a steady-state steady-flow process; the general mass balance for any control volume in rate form can be written as (Dincer and Rosen, 2007):

$$\sum_i \dot{m}_i = \sum_e \dot{m}_e \quad (3-1)$$

Where  $\dot{m}_i$  and  $\dot{m}_e$  denote the mass flow rate at specific irreversibility and specific energy respectively. The general energy and exergy balances respectively for any control volume at steady-state and steady-flow (or flow of matter) with negligible potential and kinetic energies changes through a system can be expressed as:

$$\sum_i \dot{m}_i h_i + \dot{Q} = \sum_e \dot{m}_e h_e + \dot{W} \quad (3-2)$$

and

$$\sum_i \dot{m}_i ex_i + \sum_k \dot{Ex}_{th,k} = \sum_e \dot{m}_e ex_e + \dot{Ex}^w + \dot{I} \quad (3-3)$$

Where  $\dot{Ex}_{th}$  denotes thermal (heat) exergy rate and  $\dot{Ex}^w$  exergy rate at specific work.

Note that the system exergy consumption  $\dot{I}$  is greater than zero for any irreversible process and equal to zero for a reversible process. Therefore, equation (3-3) can be written as:

$$\sum_k \dot{Ex}_{th} = \dot{Ex}^w + \dot{I} \quad (3-4)$$

and the exergy flow rate (quantity of exergy ( $Ex$ ) multiplied by the mass flow rate) of a flowing stream of matter, which can be expressed as:

$$\dot{Ex} = \dot{m}(Ex) = \dot{m}(Ex_{tm} + Ex_{ch}) \quad (3-5)$$

However, the kinetic and potential exergies are usually neglected because there is no chemical substance from the cycle to the environment, thus, the chemical exergy  $Ex_{ch}$  is zero (Dincer, 2007)(Vidal et al., 2006). The specific thermo-mechanical exergy  $ex_{tm}$  is expressed as:

$$ex_{tm} = (h - h_o) - T_o(s - s_o) \quad (3-6)$$

Usually, the analysis of a process requires the difference in thermo-mechanical exergy  $ex_{tm}$  for two states in the process rather than the reference (ambient) state (Vidal et al., 2006). Therefore, equation (3-6) can be expressed as:

$$ex_{tm} = (h_1 - h_2) - T_o(s_1 - s_2) \quad (3-7)$$

Associated with the heat transfer in and/or out of a control process from the heat source, is another form of exergy called thermal exergy  $Ex_{th}$ , which can be expressed in rate form as:

$$\dot{Ex}_{th} = \sum \left(1 - \frac{T_o}{T_{hs}}\right) \dot{Q}_{hs} \quad (3-8)$$

where  $T_{hs}$  denotes the uniform temperature of the heat source and  $\dot{Q}_{hs}$  is the heat transfer rate of the heat source.

Therefore, the exergy flow rate from equations (3-6) and (3-7) for a stream in steady flow yields:

$$\dot{Ex} = \dot{m}(h - h_o) - T_o(s - s_o) \quad (3-9)$$

Moreover, it is convenient from the described TFCs (Figures 3.1, 3.2, 3.3 and 3.4) to find the change in entropy for the  $n$ th stream  $\Delta s_n$  first and then evaluate the exergy destroyed due to thermodynamic irreversibilities during the process.

Irreversibility  $I$  is the difference between the reversible work  $W_{rev}$  and the useful work  $W_u$  due to the presence of thermodynamic irreversibilities during the process. Irreversibility is viewed as the wasted work potential or the lost opportunity to do work. The irreversibility rate  $\dot{I}$  of a process is the sum of all of available energy destruction of all the streams in the system and its rate can be shown to be equal to:

$$\dot{I} = \dot{Ex}_{dest} = T_o \sum \dot{m}_n \Delta s_n \quad (3-10)$$

The exergy analyses of the irreversibility (loss/ destruction) within a system and to the environment by each individual component in the thermodynamic loop from the TFCs schematic diagram can be expressed as (Vidal et al., 2006):

Pump:

$$\dot{I}_p = \dot{W}_p - (\dot{E}x_p^{out} - \dot{E}x_p^{in}) = N_p - \dot{m}(h_p^{out} - h_p^{in}) - T_o(s_p^{out} - s_p^{in}) \quad (3-11)$$

Where  $\dot{E}x_p^{in}$  and  $\dot{E}x_p^{out}$  denote the exergy rate at the pump inlet and outlet respectively, and  $s_p^{in}$  and  $s_p^{out}$  are the entropy at the pump inlet and outlet respectively.

Heat Exchanger:

$$\dot{I}_{hx} = \dot{E}x_{hx}^{in} - \dot{E}x_{hx}^{out} = \dot{m}(h_{hx}^{in} - h_{hx}^{out}) - T_o(s_{hx}^{in} - s_{hx}^{out}) \quad (3-12)$$

Expander:

$$\dot{I}_{exp} = (\dot{E}x_{exp}^{in} - \dot{E}x_{exp}^{out}) - \dot{W}_{exp} = \dot{m}(h_{exp}^{in} - h_{exp}^{out}) - T_o(s_{exp}^{in} - s_{exp}^{out}) \quad (3-13)$$

Condenser:

$$\dot{I}_{con} = \dot{E}x_{con}^{in} - \dot{E}x_{con}^{out} = \dot{m}(h_{con}^{in} - h_{con}^{out}) - T_o(s_{con}^{in} - s_{con}^{out}) \quad (3-14)$$

The exergy balance of the TFC can be expressed as:

$$\dot{E}x_p + \dot{E}x_{wf} = \dot{E}x_{exp} + \dot{I}_p + \dot{I}_{exp} + \dot{I}_{con} \quad (3-15)$$

Where  $\dot{E}x_p$  denotes the exergy input by the pump and  $\dot{E}x_{wf}$  is the exergy rate of the working fluid gained by absorbing heat from the heat source.

The energy balance for a system undergoing any process can be stated as ‘the net energy transfer by heat, work and mass during the process, which is equal to the change in internal, kinetic, potential, etc., energies’ (Dincer and Al-Muslim, 2001). That is,

$$E_{in} - E_{out} = E_{sys} \quad (3-16)$$

or, in the rate form, as:

$$\dot{E}_{in} - \dot{E}_{out} = dE_{sys}/dt \quad (3-17)$$

At steady-state steady-flow, the  $dE_{sys}/dt = 0$ .

Therefore,

$$\dot{E}_{in} = \dot{E}_{out} \quad (3-18)$$

Recalling that energy can be transferred by heat, work and mass only, the energy balance in equation (3-18) for a general steady-state steady-flow system with the heat input into the system at a rate of  $\dot{Q}$  and work output at a rate of  $\dot{W}$  can be expressed as:

$$\dot{Q} + \dot{m}h_1 = \dot{W} + \dot{m}h_2 \quad (3-19)$$

Since the change in kinetic and potential energies,  $\Delta ke$  and  $\Delta pe$  respectively are approximately equal to zero; i.e.,  $\Delta ke = \Delta pe \cong 0$ .

While the exergy balance for a steady-state steady-flow system undergoing any process can be stated as ‘the exergy change of a system during the process is equal to the difference between the net exergy transfer ( $Ex_{in} - Ex_{out}$ ) through the system boundary and the exergy destroyed  $Ex_{dest}$  within the system boundaries as a result of irreversibilities’ (Cengel and Boles, 2008). That is,

$$(Ex_{in} - Ex_{out}) - Ex_{dest} = \Delta Ex_{sys} \quad (3-20)$$

or, in the rate form, as:

$$\dot{Ex}_{in} - \dot{Ex}_{out} - \dot{Ex}_{dest} = dEx_{system}/dt \quad (3-21)$$

Where the rates of exergy transfer by heat and work are expressed as  $\dot{Ex}_{heat} =$

$1 - \left(T_o/T\right) \dot{Q}$  and  $\dot{Ex}_{work} = \dot{W}_{useful}$ , respectively. The exergy balance can also

be expressed per unit mass as:

$$(ex_{in} - ex_{out}) - ex_{dest} = \Delta ex_{sys} \quad (3-22)$$

where all the parameters are expressed per unit mass of the system.

However, for a reversible process, the exergy destroyed  $Ex_{dest}$  drops out from the equations (3-19) to (3-21). It is also convenient to find the entropy generation  $S_{gen}$  first and then evaluate the exergy destroyed. The exergy



destroyed is proportional to the entropy generated, which is expressed in rate form as (Cengel and Boles, 2008):

$$\dot{Ex}_{dest} = T_o \dot{S}_{gen} \quad \text{or} \quad Ex_{dest} = T_o S_{gen} \quad (3-23)$$

With specified ambient conditions, the change in exergy, which is the difference between the final and initial exergies can be determined; i.e.,

$$\Delta Ex = Ex_2 - Ex_1 = (E_2 - E_1) + P_o(V_2 - V_1) - T_o(S_2 - S_1) \quad (3-24)$$

or on per unit mass basis,

$$\Delta ex = ex_2 - ex_1 = (e_2 - e_1) + P_o(v_2 - v_1) - T_o(S_2 - S_1) \quad (3-25)$$

where  $V$  denotes the total volume of the system.

### 3.4 System Modelling

Depending on the working conditions for the described systems accompanied by the thermo-physical requirements of the TFCs, steady-state steady-flow models of the different components of the cycles, corresponding to their thermodynamic processes were established and implemented. Some standard simplifying assumptions made, based on a detailed review of existing manufacturers' catalogue, experimental rule-of-thumb and thermodynamic requirements subject to the engineering constraint imposed by the cycles are as follows:

- i. The cycles and their components operates at steady-state steady-flow conditions;

- ii. There is a thermodynamic equilibrium at the inlets and outlets of the components;
- iii. Variations of kinetic and potential energies of the heat transfer and working fluid in the cycles are negligible; and
- iv. The heat loss and pressure drop in the systems (heat exchanger and connecting pipes) are negligible.

The equipment assumptions are as follows:

- i. The expander isentropic efficiency is 90%;
- ii. The expander mechanical efficiency is 98%; and
- iii. The pump isentropic efficiency is 95%.

Using these defined parameters; i.e. the feasible operating region as specified by constraints (ii) to (iv), the performance of the cycles were analysed. The individual components of the TFCs power plants corresponding to the thermodynamic processes depicted in Figures 3.1 (b), 3.2 (b), 3.3 (b) and 3.4 (b) respectively were thermodynamically modelled.

The steady-state steady-flow models of the different components of simple TFC, corresponding to its thermodynamic processes as depicted in Figures 3.1 (b) are as follows:

For the pump,

$$\dot{W}_p = \dot{m}_{wf}(h_p^{out} - h_p^{in}) = \dot{m}_{wf}(h_{p_{is}}^{out} - h_p^{in})/\eta_p \quad (3-26)$$

For the heat exchanger,

$$\dot{Q}_{in} = \dot{m}_{wf}(h_{hx}^{out} - h_{hx}^{in}) \quad (3-27)$$

For the expander,

$$\dot{W}_{exp} = \dot{m}_{wf}(h_{exp}^{in} - h_{exp}^{out}) = \dot{m}_{wf}(h_{exp}^{in} - h_{exp\_is}^{out})\eta_{exp} \quad (3-28)$$

For the condenser,

$$\dot{Q}_{out} = \dot{m}_{wf}(h_{con}^{in} - h_{con}^{out}) \quad (3-29)$$

Where  $\dot{m}_{wf}$  denotes the mass flow rate of the working fluid,  $h_p^{in}$  and  $h_p^{out}$  are the specific enthalpy at the pump inlet and outlet respectively,  $h_{p\_is}^{out}$  is the specific enthalpy of the pump isentropic efficiency at pump outlet,  $h_{hx}^{in}$  and  $h_{hx}^{out}$  are the specific enthalpy at the heat exchanger inlet and outlet respectively,  $h_{exp}^{in}$  and  $h_{exp}^{out}$  are the specific enthalpy at the expander inlet and outlet respectively,  $h_{exp\_is}^{out}$  specific enthalpy of the expander isentropic efficiency at the expander outlet,  $h_{con}^{in}$  and  $h_{con}^{out}$  are the specific enthalpy at the condenser inlet and outlet respectively,  $\eta_p$  is the pump isentropic efficiency and  $\eta_{exp}$  is the expander isentropic efficiency.

While the steady-state steady-flow models of the different components of the TFC with IHE, corresponding to its thermodynamic processes as depicted in Figure 3.2 (b) (same as Figure 3.1 (b) except the integration of IHE), which is as follows:

For the IHE,

$$\dot{Q}_{rep} = \dot{m}_{wf}(h_{rep}^{out} - h_{rep}^{in}) \quad (3-30)$$

The steady-state steady-flow models of the different components of the reheat TFC, corresponding to its thermodynamic processes as depicted in Figure 3.2 (c) are as follows:

For the pump,

$$\dot{W}_p = \dot{m}_{wf}(h_p^{out} - h_p^{in}) = \dot{m}_{wf}(h_{p-is}^{out} - h_p^{in})/\eta_p \quad (3-31)$$

For the heat exchanger,

$$\dot{Q}_{in} = \dot{m}_{wf}(h_{hx}^{out} - h_{hx}^{in}) \quad (3-32)$$

For the high pressure expander,

$$\dot{W}_{exp, hp} = \dot{m}_{wf}(h_{exp, hp}^{in} - h_{exp, hp}^{out}) = \dot{m}_{wf}(h_{exp, hp}^{in} - h_{exp, hp-is}^{out})\eta_{exp} \quad (3-33)$$

For the reheating,

$$\dot{Q}_{rh} = \dot{m}_{wf}(h_{hx}^{out} - h_{hx}^{in}) \quad (3-34)$$

For the low pressure expander,

$$\dot{W}_{exp, lp} = \dot{m}_{wf}(h_{exp, lp}^{in} - h_{exp, lp}^{out}) = \dot{m}_{wf}(h_{exp, lp}^{in} - h_{exp, lp-is}^{out})\eta_{exp} \quad (3-35)$$

For the condenser,

$$\dot{Q}_{out} = \dot{m}_{wf}(h_{con}^{in} - h_{con}^{out}) \quad (3-36)$$

And the steady-state steady-flow models of the different components of the regenerative TFC, corresponding to its thermodynamic processes as depicted in Figure 3.3 (d) are as follows:

For the condensate pump,

$$\dot{W}_{cp} = \dot{m}_{wf}(h_{cp}^{out} - h_{cp}^{in}) = \dot{m}_{wf}(h_{cp\_is}^{out} - h_{cp}^{in})/\eta_{cp} \quad (3-37)$$

For the feed-fluid heat exchanger,

$$\dot{Q}_{fhx} = \dot{m}_{wf}(h_{fhx}^{out} - h_{fhx}^{in}) \quad (3-38)$$

For the feed pump,

$$\dot{W}_{fp} = \dot{m}_{wf}(h_{fp}^{out} - h_{fp}^{in}) = \dot{m}_{wf}(h_{fp\_is}^{out} - h_{fp}^{in})/\eta_{fp} \quad (3-39)$$

For the heat exchanger,

$$\dot{Q}_{in} = \dot{m}_{wf}(h_{hx}^{out} - h_{hx}^{in}) \quad (3-40)$$

For the expander,

$$\dot{W}_{exp} = \dot{m}_{wf}(h_{exp}^{in} - h_{exp}^{out}) = \dot{m}_{wf}(h_{exp}^{in} - h_{exp\_is}^{out})\eta_{exp} \quad (3-41)$$

For the condenser,

$$\dot{Q}_{out} = \dot{m}_{wf}(h_{con}^{in} - h_{con}^{out}) \quad (3-42)$$

The models of the cycles were established by connecting the models of individual component of the cycles and the models were employed for the rigorous performance study at the subcritical operating conditions of the cycles.

**Table 3.2:** Cycle configuration parameters

Parameters	Values [Units]
Expander isentropic efficiency	90 [%]
Pump isentropic efficiency	95 [%]
Reference temperature, $T_o$	298.15 [K]
Maximum working fluid temperature, $T_{max}$	473 [K]
Condensation temperature, $T_{con}$	309 [K]
Feed pump pressure, $P_p$	3 [MPa]

All the models process simulations and analyses of the cycles were constrained by the defined fixed cycle model input parameters presented in Table 3.2.

### 3.4.1 Thermodynamic and Design Methodology

Generally, the process simulation models always require numerous model input parameters and defining those that might easily be distinguished for a significant effect on performance metrics (measures or responses) are overwhelming undertaking. The typical approach of varying single parameter at a time is often inappropriate. This is due to the fact that numerous model parameters act together to influence response.

In this study, a thermodynamic approach is used for the design of simulation models of the cycles, which is aimed at establishing simplified thermodynamic

models to predict performance response or to determine the combination of parameters that optimizes performance metrics of the cycles.

### **3.4.2 Simulation Algorithm**

The operating criteria of the simulation program are specifically complex. Firstly, subprograms were particularly established for individual components of the power plants in EES environment. All subprograms consist of input and output parameters and specified equations, which permit the computation of the performance metrics of the cycles based on the fixed input model parameters. Distinctly, all the subprograms of each cycle are precisely interconnected and they embody the realistic physics of interconnections, which transpire among the components of the individual cycles.

### **3.4.3 Computation Methodology**

In the heat exchanger, the temperature and the mass flow rate of the heat source were fixed as well as the maximum temperature of the working fluid. With the energy balance of the heat exchanger, the return working fluid temperature is initialised; consequently working fluid mass flow rate and the heat source temperature were computed.

With a fixed expander inlet pressure at the average condensing temperature, the working fluid temperature and its mass flow rate were computed. At the condenser, the expander exhaust (or discharge) pressure can be computed by exploiting the saturated temperature of the working fluid. Thus, the condensing pressure was assessed using the ambient temperature. By setting the

condenser's cooling system temperature differences, the cooling fluid mass flow rate was calculated.

Using the subcritical operating inlet conditions of the expander, expander exhaust pressure and its isentropic efficiency, the expander's specific work output and the exhaust conditions of the working fluid can be assessed. Thus, net work output was computed based on the product of the mass flow rate of the working fluid and the expander's specific work output.

Consequently, with the defined fixed values of the model input parameters based on the thermodynamic and design basis methodology, standard process simulation runs of the models were performed at the subcritical operating conditions. Using the defined key input parameters; an expander inlet pressure of 3 MPa, expander inlet temperatures of 473 K and expander isentropic efficiency of 90% at the average condensing temperature of 309 K; the various cycles' thermodynamic states were computed as well as the performance metrics of the cycles. The working fluid at the expander exit state was assessed to safeguard that its mostly dry (90% or above minimum dryness fraction) at the exhaust of the expander for all simulations performed.

The thermal efficiency, exergy efficiency and net work output of the cycles were evaluated by varying the expander inlet pressure over 2 - 3 MPa and the expander inlet temperature over 393 - 473 K at the average condensing temperature of 309 K.



### 3.5 Thermodynamic Analysis

Thermal efficiency, which is dependent on the first-law of thermodynamics, is typically utilized to assess and compare power cycles. The thermal efficiency of the simple TFC  $\eta_{th}$  is:

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{\dot{W}_{exp} - \dot{W}_p}{\dot{Q}_{in}} \quad (3-43)$$

The thermal efficiency of the TFC with IHE is expressed as:

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{\dot{W}_{exp} - \dot{W}_p}{\dot{Q}_{in}} = \frac{\dot{W}_{exp} - (\dot{W}_{cp} + \dot{W}_{fp})}{\dot{Q}_{hx} + \dot{Q}_{rep}} \quad (3-44)$$

The thermal efficiency of the TFC with reheating is expressed as:

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{\dot{W}_{exp,hp} + \dot{W}_{exp,lp} - \dot{W}_p}{\dot{Q}_{hx} + \dot{Q}_{rh}} \quad (3-45)$$

And the thermal efficiency of the regenerative TFC is expressed as:

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{\dot{W}_{exp} - \dot{W}_p}{\dot{Q}_{in}} = \frac{\dot{W}_{exp} - (\dot{W}_{cp} + \dot{W}_{fp})}{\dot{Q}_{hx} + \dot{Q}_{fhx}} \quad (3-46)$$

The exergy (second-law) efficiency  $\eta_{II}$  of a system during a thermodynamic process (or of various steady-flow devices) is defined as (Cengel and Boles, 2008):

$$\eta_{II} = \frac{\text{Exergy recovered}}{\text{Exergy supplied}} = 1 - \frac{\text{Exergy destroyed}}{\text{Exergy supplied}} \quad (3-47)$$

The exergy efficiency for a power cycle can be defined as the ‘ratio of the actual thermal efficiency to the maximum possible (reversible) thermal efficiency under same conditions’ (Cengel and Boles, 2008). That is,

$$\eta_{II} = \frac{\eta_{th}}{\eta_{th,rev}} \quad (3-48)$$

and reversible thermal efficiency  $\eta_{th,rev}$  (Carnot equivalent) can be determined as follows:

$$\eta_{th,rev} = \left(1 - \frac{T_c}{T_h}\right) \quad (3-49)$$

Where  $T_c$  and  $T_h$  denote the temperatures of heat sink (or cold stream) and heat source (or hot streams) respectively.

For work producing devices (e.g. expanders), the exergy efficiency can also be expressed as ‘the ratio of the useful work output and the maximum possible (reversible) work output’. That is,

$$\eta_{II} = \frac{W_u}{W_{rev,max}} \quad (3-50)$$

While for work consuming devices (e.g. pumps), the exergy efficiency can also be expressed as ‘the ratio of the minimum (reversible) work input to the useful work input’. That is,

$$\eta_{II} = \frac{W_{rev,min}}{W_u} \quad (3-51)$$

### 3.6 Results and Discussion

The trilateral flash cycles (TFCs) power plants that comprised a heat source, a power cycle and a heat sink; employing *n*-pentane as the working fluid were thermodynamically modelled, implemented and analysed. The process simulation runs of *n*-pentane based simple TFC, TFC with internal heat exchanger (IHE), reheat TFC and regenerative TFC and their computed results based on the expander inlet pressure of 3 MPa, expander inlet temperature of 473 K and expander isentropic efficiency of 90% at the average condensing temperature of 309 K were presented ([Appendices A.1 to A.4](#)). The cycles' thermodynamic performances were analysed by varying the expander inlet pressure over 2 - 3 MPa and expander inlet temperature over 393 - 473 K.

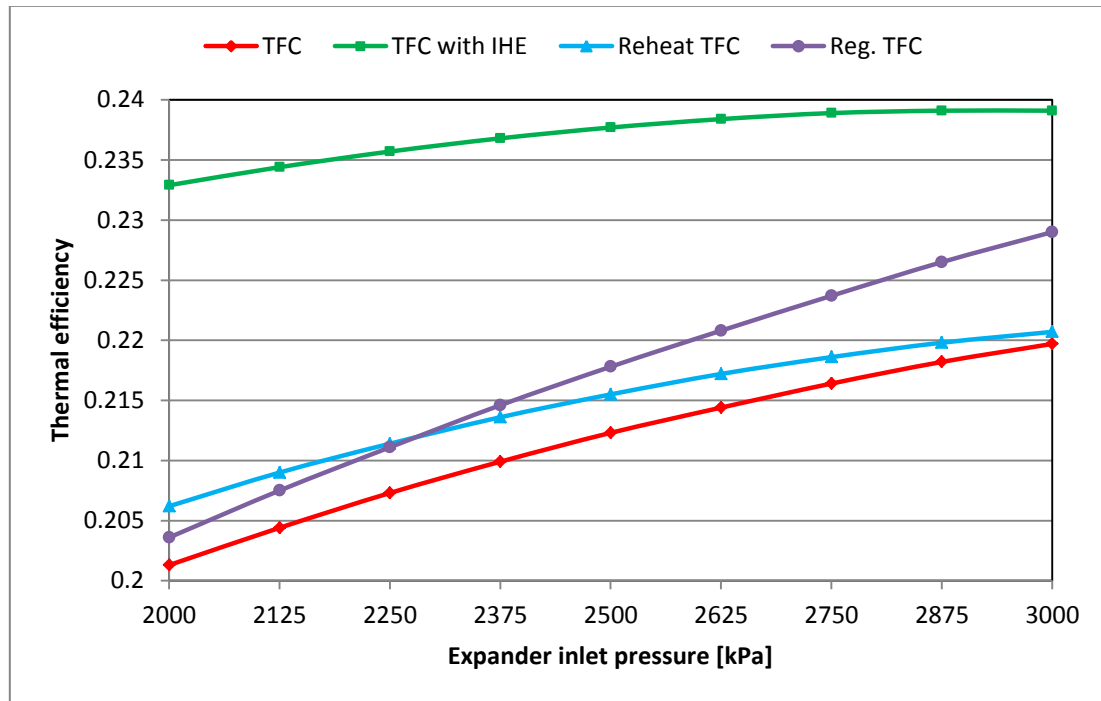
Table [3.3](#) listed the key set of the expander work outputs and performance metrics of the cycles for the design case based on the expander inlet pressure of 3 MPa, expander inlet temperature of 473 K and expander isentropic efficiency of 90% at the average condensing temperature of 309 K.

**Table 3.3:** Key cycles work outputs simulation results and their performance metrics

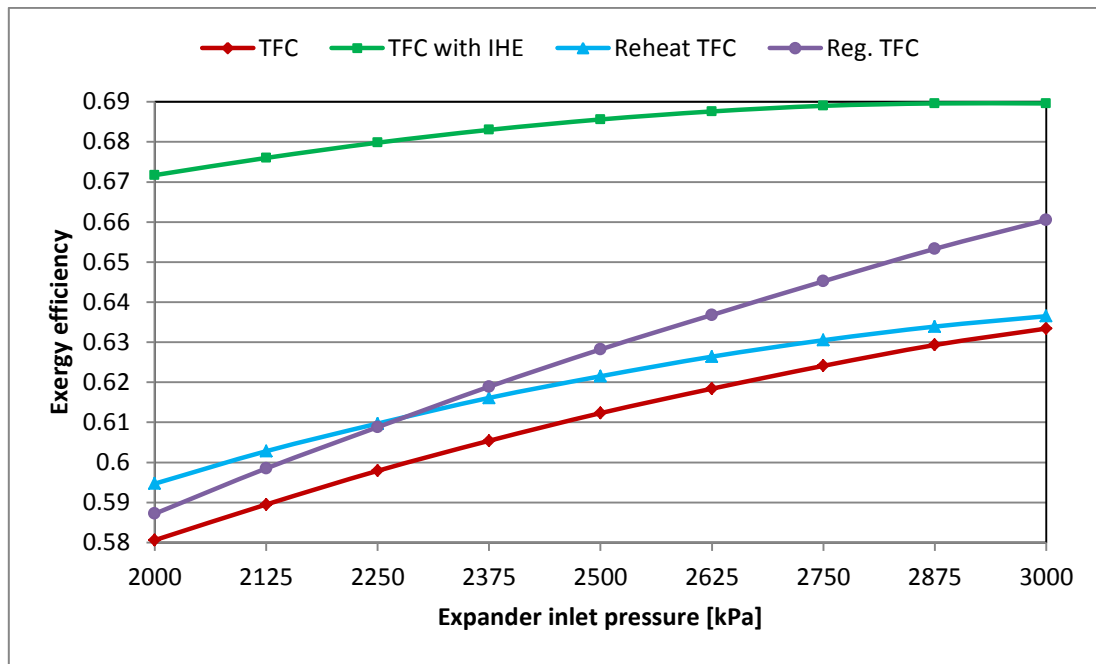
Simple TFC	<b>Power output result</b>	<b>Design basis values [Units]</b>
	Expander work output	139.1 [kW]
	<b>Performance metrics</b>	<b>Design basis values [Units]</b>
	Thermal efficiency	21.97 [%]
	Exergy efficiency	63.36 [%]
Simple TFC with IHE	Net work output	134.1 [kW]
	<b>Power output result</b>	<b>Design basis values [Units]</b>
	Expander work output	150.9 [kW]
	<b>Performance metrics</b>	<b>Design basis values [Units]</b>
	Thermal efficiency	23.91 [%]
Reheat TFC	Exergy efficiency	68.96 [%]
	Net work output	145.9 [kW]
	<b>Power output result</b>	<b>Design basis values [Units]</b>
	Expander work output	129.07 [kW]
	<b>Performance metrics</b>	<b>Design basis values [Units]</b>
Regenerative TFC	Thermal efficiency	22.07 [%]
	Exergy efficiency	63.65 [%]
	Net work output	124.1 [kW]
	<b>Power output result</b>	<b>Design basis values [Units]</b>
	Expander work output	135.82 [kW]
Regenerative TFC	<b>Performance metrics</b>	<b>Design basis values [Units]</b>
	Thermal efficiency	22.9 [%]
	Exergy efficiency	66.05 [%]
	Net work output	130.5 [kW]

The results of performance variation with expander inlet pressure are given in Figure 3.6. Figure 3.6 (a) depicts the variation of thermal efficiencies of the various cycles with expander inlet pressure. The resulting thermal efficiencies of the cycles increased as their inlet pressure increased from 2 - 3 MPa. This is

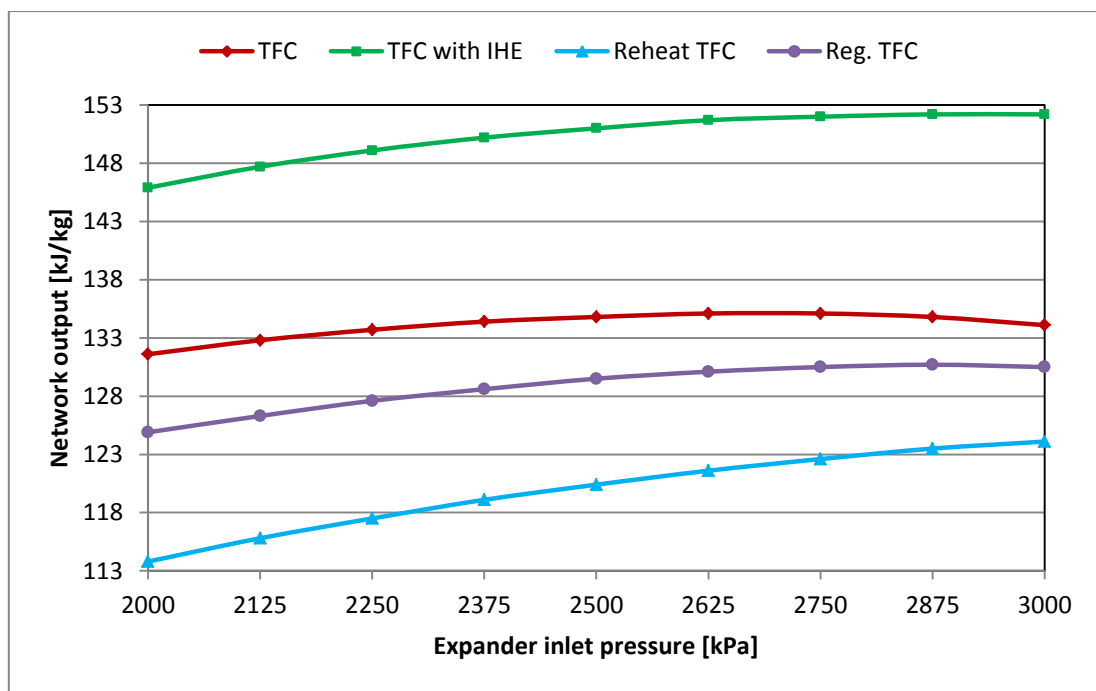
because a meaningful well-pressurized working fluid was injected into the expander at higher pressure per unit mass flow rate of the working, while the expander outlet pressure remained nearly constant. The observed incremental rises in thermal efficiencies diminished in magnitudes as the expander inlet pressure increased. Figure 3.6 (b) depicts the variation of exergy efficiencies of the cycles with inlet pressure, showing how their exergy efficiencies increased as the inlet pressure increased from 2 - 3 MPa.



(a)



(b)



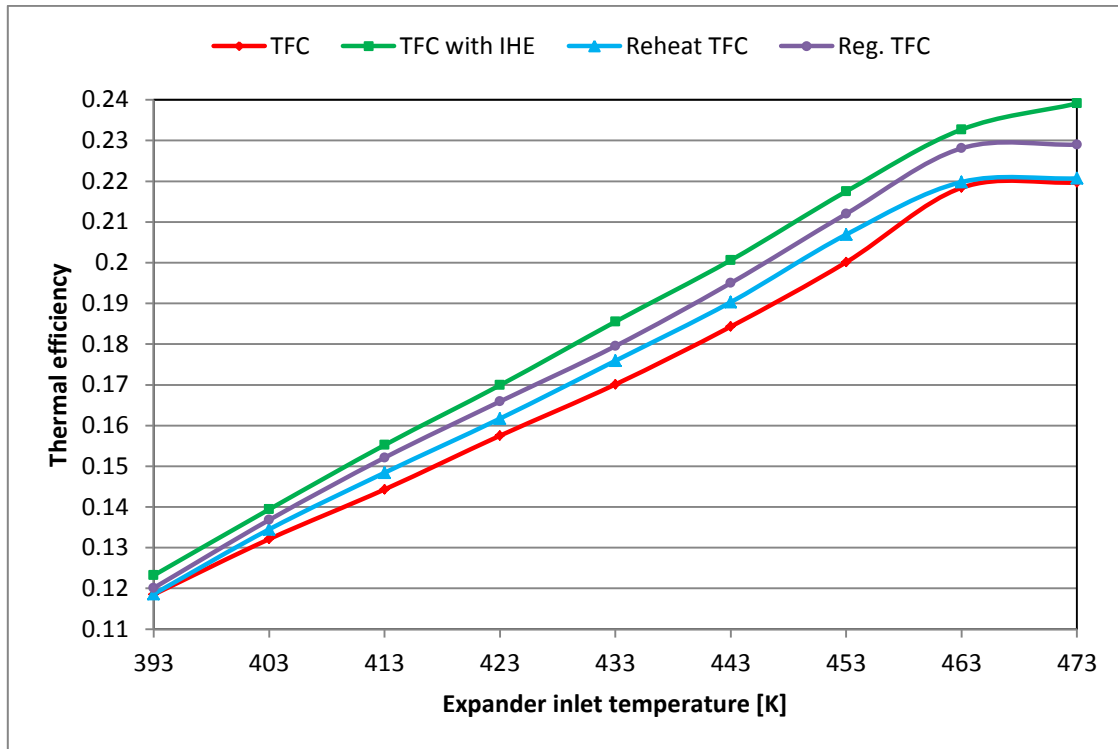
(c)

**Figure 3.6:** Performance variation with expander inlet pressure

Figure 3.6 (c) depicts the variation of net work outputs of the cycles with inlet pressure, showing how their net work output increased as the inlet pressure increased from 2 - 3 MPa. The observed behavioural trends are exciting as the net work outputs of the cycles increased with increasing inlet pressure until their respective maximal were attained (at 'different pressure levels') and afterward diminished as the pressure increased.

The results of thermal efficiency variation with expander inlet temperature are given in Figure 3.7. Figure 3.7 depicts the variation of thermal efficiencies of the various cycles with expander inlet temperature. The resulting thermal efficiencies of the cycles increased as their inlet temperature increased from 393 - 473 K. This is because more quantities of thermal energy per unit mass of the working fluids were injected into the expander at higher temperature, while the expander outlet (or condensing) temperatures remain nearly constant. The observed incremental rises in thermal efficiencies diminished in magnitudes as the expander inlet temperature increased.

In conclusion, the effects of varying expander inlet pressure and temperature on the absolute levels attained by performance metrics of the cycles were significant. As inlet pressure increases, optimal pressure ratios between the expander inlets and their outlets increased, resulting in higher efficiencies due to increased net work outputs from the cycles. While as inlet temperature increases, a meaningful more thermal energy per unit mass of the working fluid were injected into the expander, resulting in higher system efficiencies.



**Figure 3.7:** Performance variation with expander inlet temperature

### 3.7 Conclusions

This study presents modelling, simulation and evaluation of the trilateral flash cycles (TFCs) power plants employing *n*-pentane as the working fluid for low-grade waste heat recovery-to-power generation. Steady-state steady-flow models of TFCs were modelled and implemented. These models were used to determine the optimum synthesis/ design parameters of the cycles and to evaluate their energetic performance, at the subcritical operating conditions and design criteria. Thus, they might be valuable tools in the preliminary prototype system design of the power plants.

A system design criteria was established subject to engineering design constraints and the corresponding thermodynamic processes of the simple



TFC, TFC with internal heat exchanger (IHE), reheat TFC and regenerative TFC were thermodynamically modelled and implemented. These were used to examine the performance metrics of the cycles at the subcritical operating conditions with low-grade heat in the range of 393 K - 473 K and the design criteria. The results depict that the thermal and exergy efficiencies, and as well the net work outputs increased as the expander inlet pressures increased from 2 - 3 kPa (Figure 3.6 (a) - (c)). Similarly, the thermal efficiency increased as the expander inlet temperature increased from 393 - 473 K (Figure 3.7). The thermal efficiencies of the cycles increased from 11.85 - 21.97%, 12.32 - 23.91%, 11.86 - 22.07% and 12.01 - 22.9% for the simple TFC, TFC with internal heat exchanger, reheat TFC and regenerative TFC respectively. While the plants were found to generate 134.1 kW, 145.9 kW, 124.1kW and 130.5 kW of power for the simple TFC, TFC with internal heat exchanger, reheat TFC and regenerative TFC respectively.

**CHAPTER FOUR**

**THERMODYNAMIC STUDY OF TRILATERAL FLASH  
CYCLES**

## 4 Introduction

At present, heat recovery-to-power technologies have been resourcefully exploited for sustainable power generation; bridging the growing gap between energy needs and its sustainable and affordable supply, improving the overall energy efficiency and reducing the unit costs of energy (Heppenstall, 1998; Demirkaya et al., 2011). Consequently, decreasing fossil fuels burning and relaxing greenhouse gases emission (Smith et al., 2005) and thermal pollution of the environment (Dai et al., 2009b).

Conventionally, the steam Rankine cycle has been broadly utilized for heat recovery-to-power generation from thermal energy of external heat sources, such as combustion products (Smith et al., 2011). However, due to the low conversion efficiency (Franco and Casarosa, 2002; Chen et al., 2011a) or cost of the exploiting available work (Chan et al., 2013), low-grade heat streams of numerous renewable energies and exhaust waste heat sources are ordinarily under-exploited.

The consequence of these, are the growing interest in the quest for innovative low-grade heat recovery-to-power technologies (Zamfirescu and Dincer, 2008; Zamfirescu and Dincer, 2009). Among them, the trilateral flash cycle (TFC) is a promising heat recovery-to-power cycle, which presents a great potential for development. Few studies and developmental efforts from the proof-of-concept towards a market-ready technology of this promising cycle have been carried out to demonstrate its feasibility for heat recovery-to-power generation onto the grid as well as for off-grid electricity supply.

In contrast to the Rankine cycles, which use the conventional working fluid: the vapour steam; innovative heat recovery-to-power cycles use unconventional working fluids: refrigerants or hydrocarbons (Tchanche et al., 2009). The economics of these cycles depend on the thermo-physical properties of the working fluid and the consequence of a wrong choice would be an inefficiently low and costly power plant (Andersen and Bruno, 2005; Tchanche et al., 2009). Thus, in a thermodynamic cycle, the working fluid performs a vital role in the processes of the cycle (Chen et al., 2010), and a careful selection that meets the cycle requirements is crucial for an 'efficient and safe operation' (Hung et al., 2010). Its applicability range must be within its thermo-physical properties and the chemical stability in a desirable range of temperature (Chen et al., 2010). Moreover, the operating conditions of the thermodynamic cycle, its system efficiency, economic feasibility and environmental impact are influenced by its fluid selection (Liu et al., 2004; Desai and Bandyopadhyay, 2009; Rayegan and Tao, 2011).

There are several integration techniques for efficient usage of thermal energy like the pinch technology (El-Sayed, 2003; Seider et al., 2004), thermo-economic analysis (Uran, 2006; Gassner and Maréchal, 2008; Quoilin et al., 2011), exergy analysis (Rivero, 2001; Dagdas, 2005; Cihann et al., 2006; Dai et al., 2009a) and exergo-economic analysis (Hua et al., 1997; Kwon et al., 2001; Cardona and Piacentino, 2006) that have been employed to analyse, evaluate and compare different power cycles in the recent times.

Exergy analysis is universally acceptable in the efficiency analysis of energy systems and other processes or systems. This is because it is a systematic

technique that allows localization and the measure of the degree of inefficiency, depicting the utmost inefficient components of a process or system (Brodyanski et al., 1994). For heat recovery-to-power generation systems, the exergy-based assessment permits the determination of the maximum potential for power generation depending on the input thermal flows into the system. This maximum is achievable only when the utilization of the input thermal flow into the processes ultimately brings it to complete thermodynamic equilibrium with the environment, while generating power.

#### **4.1 Trilateral Flash Cycles**

The simple trilateral flash cycle (TFC) power plant was studied for low-grade waste heat recovery-to-power generation. In order to improve the heat exchange performance and system efficiency, the TFC was integrated with an internal heat exchanger (IHE), reheating and feed fluid-heating. Thus, four distinct system configurations of the TFC: simple TFC, TFC with IHE, reheat TFC and TFC with feed fluid-heating (or regenerative TFC) were thermodynamically analysed.

Previously established and implemented steady-state steady-flow models of the cycles were used to thermodynamically assess the performance metrics of the cycles and their predictions. The models were established based on the corresponding thermodynamic processes of the power cycles. The thermodynamic properties for temperature, pressure, enthalpy and entropy as well as the performance of the cycles during process simulations were predicted at the subcritical operating conditions of the cycle. The results were analysed

for all cycles and their operating characteristics were established within the viable configuration (working) parameters.

A detailed thermodynamic analysis was performed using an expander inlet pressure of 3 MPa and expander isentropic efficiency of 90% at the cycle high temperature of 473 K and average condensing temperature of 309 K. The thermal and exergy efficiencies of all the cycles as well as their exergy efficiencies of the condensing and heating processes were computed at the cycle high temperature of 473 K and average condensing temperature of 309 K.

#### **4.1.1 Descriptions of Trilateral Flash Cycle Systems**

The schematic cycle configurations of the four TFC power plants are shown in Figure 4.1, illustrating the distinct operating features of the various system configuration designs.

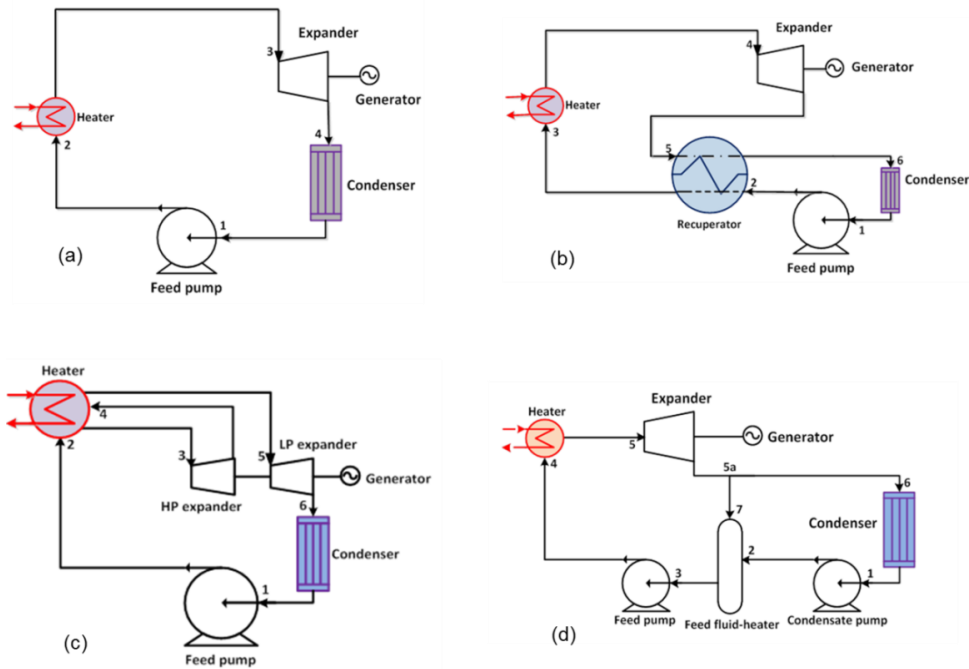
Figure 4.1 (a) depicts the schematic cycle configuration of the simple TFC, which comprises four key components: feed pump, heater, expander and condenser. Similar to the Rankine cycles, the saturated working fluid is pressurized (state 1 - 2) to a higher pressure, afterward heated just to its saturated temperature (state 2 - 3) at constant pressure and is being injected into the expander (state 3 - 4); where shaft work is produced, driving a generator to produce power. Subsequently, the resulting vapour–liquid content is condensed (state 4 - 1) to start the new cycle.

Figure 4.1 (b) depicts the schematic cycle configuration of the TFC with IHE, which comprises five key components: feed pump, heater, expander, IHE and

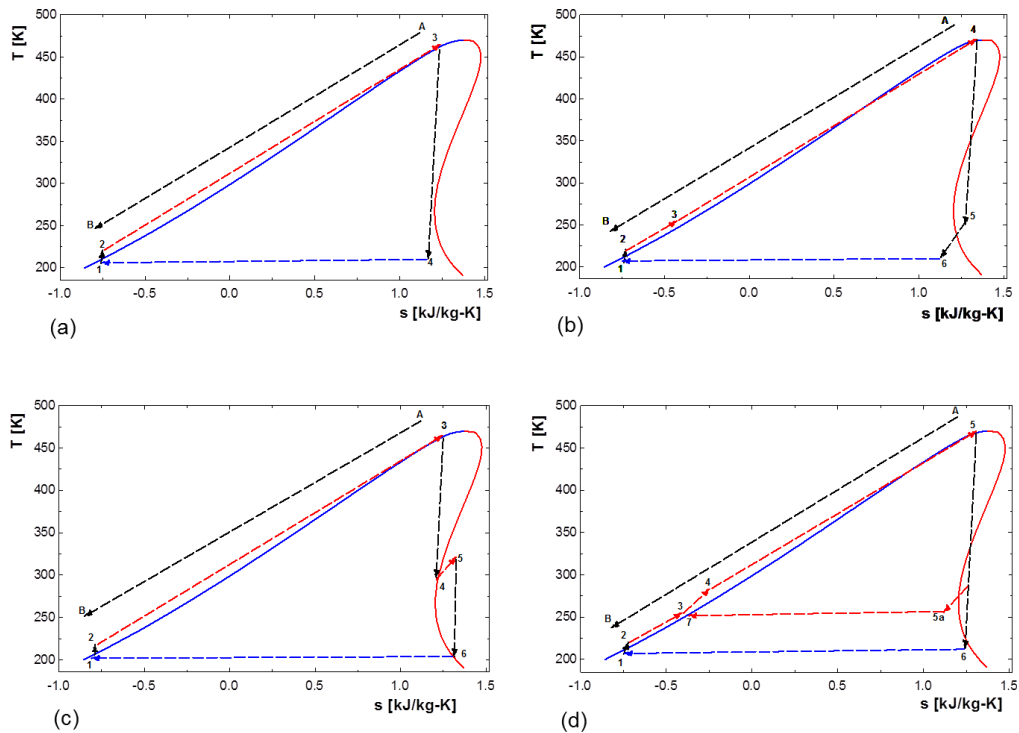
condenser. The latent heat extracted by the IHE at the expander outlet is used to preheat the sub-cooled liquid at the pump outlet.

Figure 4.1 (c) depicts the schematic cycle configuration of the reheat TFC, which comprises five key components: feed pump, heater, high pressure (HP) expander, low pressure (LP) expander and condenser. Unlike the simple TFC, the high pressure fluid is injected into the HP expander (state 3 - 4); where it is first expanded. Afterwards, the fluid is then returned to the heater where it is reheated to its saturated temperature (state 4 - 5) and is expanded in the low LP expander (state 5 - 6), with the resulting vapour–liquid content then condensed (state 6 - 1) to start the new cycle.

Figure 4.1 (d) depicts the schematic cycle configuration of the regenerative TFC, which comprises six key components: condensate pump, feed fluid-heater, feed pump, heater, expander and condenser. The working fluid is pressurized by the condensate pump (state 1 - 2), afterwards preheated in the feed fluid-heater (state 2 - 3) and is being pressurized to a high pressure with the feed pump (state 3 - 4). The high pressure fluid is heated (state 4 - 5) and injected into the expander (state 5 - 6); where shaft work is derived, with the resulting vapour–liquid content being bled (state 5a) for preheating the sub-cooled liquid at the condensate pump outlet (process 5a - 7) and condensed (state 6 - 1).



**Figure 4.1:** The trilateral flash cycle, showing the schematic cycle configurations of the simple TFC in (a), TFC with IHE in (b), reheat TFC in (c) and regenerative TFC in (d)



**Figure 4.2:** The T-s diagrams of the trilateral flash cycles, showing the thermodynamic process of the simple TFC in (a), TFC with IHE in (b), reheat TFC in (c) and regenerative TFC in (d)



The cycles' temperature–entropy ( $T$ – $s$ ) diagrams of their thermodynamic processes are shown in Figure 4.2. The simple TFC comprises four internally reversible processes. These include processes 1 - 2 and 3 - 4 that depict the isentropic compression and expansion processes of the working fluid respectively during which work is either performed on or produced from the cycle; and processes 2 - 3 and 4 - 1 that depict the constant pressure high-temperature heat addition and low-temperature heat rejection processes of the working fluid (Figure 4.2 (a)). While states 3 and 5 in Figure 4.2 (b) depict the integration of IHE in the simple TFC; the reheating process is depicted by states 4 and 5 (Figure 4.2 (c)) and states 5a and 7 (Figure 4.2 (d)) depicted the regeneration process.

#### 4.1.2 *n*-Pentane

The working fluid adopted for the study is *n*-pentane because of its good thermo-physical properties (e.g. relative high critical temperature and pressure), low-cost, and a boiling point slightly above room temperature. It displays a strong positive slope on the  $T$ – $s$  diagram and its saturated liquid expansion tends to dry out at temperatures slightly exceeding 453 K (Smith and da Silva, 1994). More so, *n*-pentane is a dry fluid (Desai and Bandyopadhyay, 2009), whose thermo-physical properties are well-suited for low-grade heat recovery-to-power generation.

## 4.2 System Modelling

The steady-state steady-flow process simulation models of the described systems, corresponding to their thermodynamic processes were

thermodynamically established and implemented employing EES (engineering equation solver) (Klein, 2013). The models process simulations with defined fixed cycle configuration (or model) input parameters at the subcritical operating conditions were carried out and results obtained.

The results of the process simulations of the *n*-pentane based simple TFC, TFC with IHE, reheat TFC and regenerative TFC were computed based on the inlet pressure of 3 MPa and expander isentropic efficiency 90% at the cycle high temperature of 473 K and average condensing temperature of 309 K.

### 4.3 Thermodynamic Analysis

The thermodynamic analysis for all the cycles was evaluated at the cycle high temperature of 473 K and average condensing temperature of 309 K.

#### 4.3.1 Energy and Exergy Efficiencies of the Trilateral Flash Cycles

Thermal efficiency, which is dependent on the first-law of thermodynamics, is typically utilized to assess and compare power cycles. The first-law (energy or thermal) efficiency  $\eta_{th}$  is basically expressed as the 'ratio of the useful energy output to the total energy input'. Mathematically, it is expressible for simple TFC system as:

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{\dot{W}_{exp} - \dot{W}_p}{\dot{Q}_{in}} \quad (4-1)$$

While the thermal efficiency of the TFC with IHE is expressed as:

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{\dot{W}_{exp} - \dot{W}_p}{\dot{Q}_{in}} = \frac{\dot{W}_{exp} - (\dot{W}_{cp} + \dot{W}_{fp})}{\dot{Q}_{hx} + \dot{Q}_{rep}} \quad (4-2)$$

The thermal efficiency of the TFC with reheating is expressed as:

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{\dot{W}_{exp,hp} + \dot{W}_{exp,lp} - \dot{W}_p}{\dot{Q}_{hx} + \dot{Q}_{rh}} \quad (4-3)$$

And the thermal efficiency of the regenerative TFC is expressed as:

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{\dot{W}_{exp} - \dot{W}_p}{\dot{Q}_{in}} = \frac{\dot{W}_{exp} - (\dot{W}_{cp} + \dot{W}_{fp})}{\dot{Q}_{hx} + \dot{Q}_{fhx}} \quad (4-4)$$

In a thermodynamic cycle, total exergy input comprises the work input by the feed pump and as well the working fluid's exergy input by thermal exergy transferred from the heat source; while net work output is the exergy output of the cycle. Therefore, the exergy efficiency of a power cycle  $\eta_{II,cyc}$  can be expressed as (Chen et al., 2011b):

$$\eta_{II,cyc} = \frac{\dot{W}_{net}}{\dot{W}_p + \Delta \dot{E}x_{wf}} \quad (4-5)$$

Where  $\Delta \dot{E}x_{wf}$  denotes the working fluid's exergy acquired by the heat exchange with the heat source, which is:

$$\Delta \dot{E}x_{wf} = \dot{m}_{wf} [(h_{wf}^{in} - h_{wf}^{out}) - T_o (s_{wf}^{in} - s_{wf}^{out})] \quad (4-6)$$

### 4.3.2 Exergy Efficiency of Heat Exchange

As the heat exchange (or heat recovery) takes place from a high-temperature (or hot streams) to a low-temperature (or cold streams) irreversibly, 'exergy is destroyed'. However, the rate of irreversibility (exergy destruction) per unit thermal exergy transferred is substantially low, if the hot and cold streams are substantively higher than the ambient temperature (Chen, 2010). This suggests that maximizing the thermal match (or heat recovery) in the condensing process is advantageous than in heating.

For a heat transfer process, i.e. the heating and condensing processes, the change of exergy flow of the hot stream is:

$$\Delta \dot{E}x_h = \dot{m}_h [(h_h^{in} - h_h^{out}) - T_o(s_h^{in} - s_h^{out})] \quad (4-7)$$

and the exergy change of the cold stream is:

$$\Delta \dot{E}x_c = \dot{m}_c [(h_c^{in} - h_c^{out}) - T_o(s_c^{in} - s_c^{out})] \quad (4-8)$$

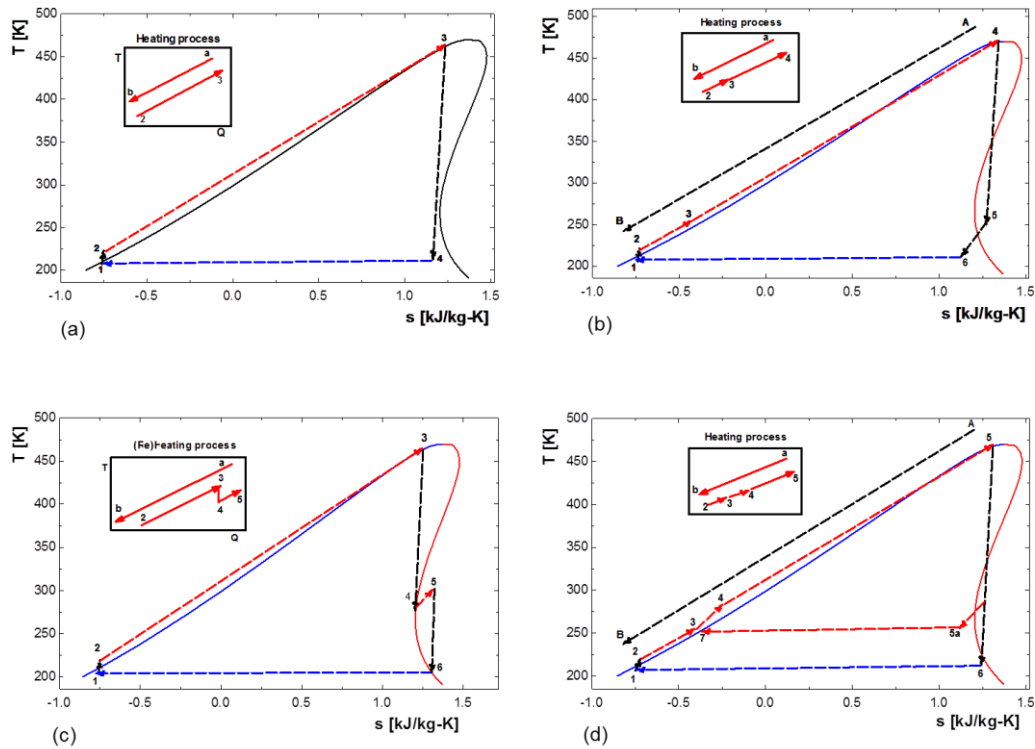
Therefore, the heat exchange exergy efficiency  $\eta_{II,hex}$  is expressed as (Chen, 2010):

$$\eta_{II,hex} = \frac{\Delta \dot{E}x_{cp}}{\Delta \dot{E}x_{hp}} = \frac{\Delta \dot{E}x_c}{\Delta \dot{E}x_h} = \frac{\dot{m}_c [(h_c^{in} - h_c^{out}) - T_o(s_c^{in} - s_c^{out})]}{\dot{m}_h [(h_h^{in} - h_h^{out}) - T_o(s_h^{in} - s_h^{out})]} \quad (4-9)$$

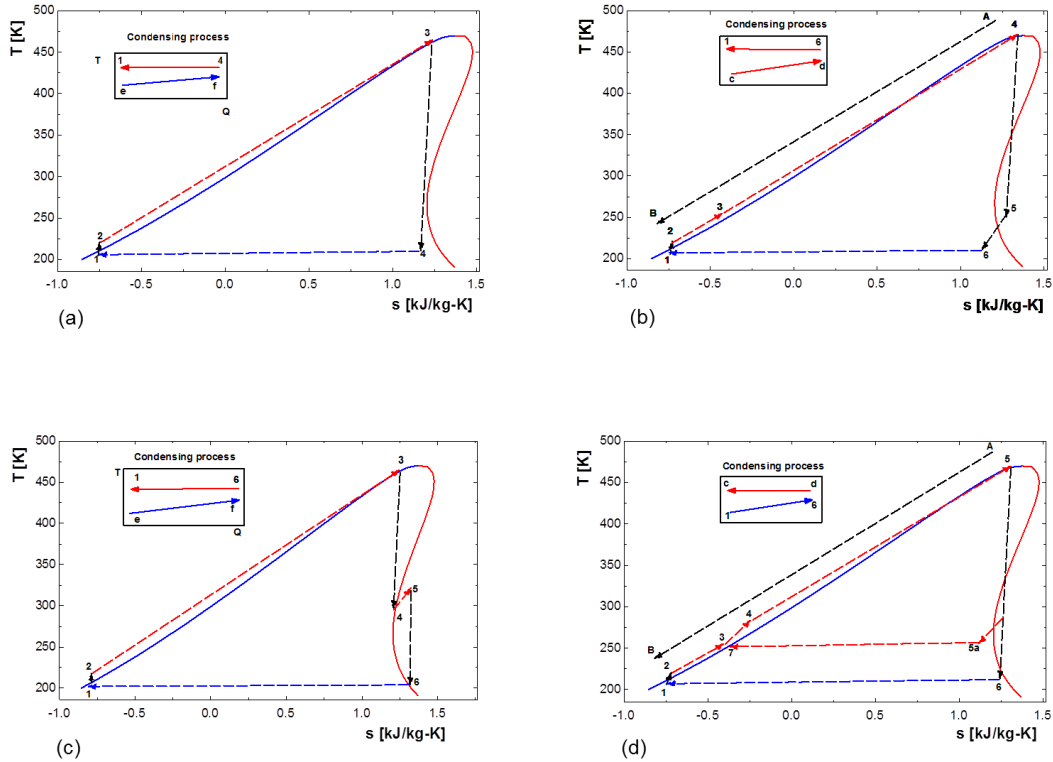
#### 4.3.2.1 Exergy Efficiency of Heating and Condensing Processes

The exergy analysis of the cycles was used to examine the performance of the heating (i.e. the heat transfer to the cycle from the heat source) and condensing

(i.e. the heat rejection by the cycle to the heat sink) processes of the  $n$ -pentane in the cycles. Figures 4.3 (a) - (d) and Figures 4.4 (a) - (d) are the T–s diagrams of the thermodynamic processes of the cycles, illustrating their thermal match of the heating and condensing processes at the top left corners of the figures.



**Figure 4.3:** Heating process of the  $n$ -pentane in the cycles and their thermal match with the heat source



**Figure 4.4:** Condensing process of the *n*-pentane in the cycles and their thermal match with the cooling fluid

Process simulations of the cycles heating processes were carried out using heat transferred to the working fluid at state 2 to 3 (Figures 4.3 (a), (b) and (d)) and at state 2 to 3 alongside state 4 to 5 (Figure 4.3 (c)) by a pressurized heat source of 610.4 kJ/kg (0.5 MPa). The mass flow rate of the working fluid for each cycle (in  $\text{kgs}^{-1}$ ) is heated to cycle high temperature of 473 K, assuming a fixed mass flow rate of heat source and its temperature that is constantly sufficient for heating the working fluid to 473 K. While those of the condensing processes were carried out using heat dissipated to the cooling fluid by the working fluid from saturated vapour–liquid condition to saturated liquid.

The exergy flow from the heat source at the heat exchanger inlet (point a) to the outlet (point b) (Figures 4.3 (a), (b) and (d)) of the cycles are:

$$\Delta \dot{E}x_{hs} = \dot{m}_{hs}(\dot{e}_a - \dot{e}_b) = \dot{m}_{hs}[(h_a - h_b) - T_o(s_a - s_b)] = 0 \quad (4-10)$$

and the exergy flow rate to the working fluid from subcooled liquid (state 2) to saturated liquid temperature (state 3) (Figures 4.3 (a), (b) and (d)) is similarly expressed as:

$$\Delta \dot{E}x_{wf} = \dot{m}_{wf}(\dot{e}_3 - \dot{e}_2) = \dot{m}_{wf}[(h_3 - h_2) - T_o(s_3 - s_2)] = 0 \quad (4-11)$$

Where  $\dot{e}_a - \dot{e}_b$  and  $\dot{e}_3 - \dot{e}_2$  denote the changes in energy flow per unit mass from point *a* to point *b* and from state 2 to state 3 respectively,  $\dot{m}_{hs}$  is the heat source mass flow rate,  $h_a$  and  $h_b$  are the heat source specific enthalpies at points *a* and *b* respectively and  $h_2$  and  $h_3$  are the working fluid specific enthalpies at states 2 and 3 respectively.

Hence, the exergy efficiency of the heating process  $\eta_{II, hp}$  of the working fluid for the cycles were computed to be 86.15%, 92.33% and 88.74% for the simple TFC, TFC with IHE and regenerative TFC respectively using Eq. (4.12):

$$\eta_{II, hp} = \frac{\Delta \dot{E}x_{hs}}{\dot{E}x_{wf}} = \frac{\dot{m}_{hs}(\dot{e}_a - \dot{e}_b)}{\dot{m}_{wf}(\dot{e}_3 - \dot{e}_2)} \quad (4-12)$$

While the exergy flow from the heat source at the heat exchanger inlet (point *a*) to the outlet (point *b*) coupled with reheating at point *c* to *d* (Figure 4.4 (c)) of the reheat TFC is expressed as:

$$\begin{aligned}
\Delta \dot{E}x_{hs} &= \dot{m}_{hs}((\dot{e}_a - \dot{e}_b) + (\dot{e}_c - \dot{e}_d)) \\
&= \dot{m}_{hs}[(h_a - h_b) + (h_c - h_d) - T_o((s_a - s_b) + (s_c - s_d))] \\
&= 0
\end{aligned} \tag{4-13}$$

and the exergy flow rate from the subcooled liquid (state 2) to saturated liquid temperature (state 3) coupled with the flow from the reheated resulting vapour–liquid (state 4) to saturated liquid temperature (state 5) (Figure 4.4 (c)) is similarly expressed as:

$$\begin{aligned}
\Delta \dot{E}x_{wf} &= \dot{m}_{wf}((\dot{e}_3 - \dot{e}_2) + (\dot{e}_5 - \dot{e}_4)) \\
&= \dot{m}_{wf}[(h_3 - h_2) + (h_5 - h_4) - T_o((s_3 - s_2) + (s_5 - s_4))] \\
&= 0
\end{aligned} \tag{4-14}$$

Hence, the exergy efficiency of the heating process  $\eta_{II, hp}$  of the working fluid for the reheat TFC was computed to be 87.45% using Eq. (4.15):

$$\eta_{II, hp} = \frac{\Delta \dot{E}x_{hs}}{\Delta \dot{E}x_{wf}} = \frac{\dot{m}_{hs}((\dot{e}_a - \dot{e}_b) + (\dot{e}_c - \dot{e}_d))}{\dot{m}_{wf}((\dot{e}_3 - \dot{e}_2) + (\dot{e}_5 - \dot{e}_4))} \tag{4-15}$$

While the mass flow rate of the cooling fluid is expressed based on the energy balance of the condenser at steady-state steady-flow as:

$$\dot{m}_{wat}(h_f - h_e) + \dot{m}_{wf}(h_4 - h_1) = 0 \tag{4-16}$$

Hence, the cooling fluid mass flow rate  $\dot{m}_{wf}$  was computed to be 27.77 kgs<sup>-1</sup> using Eq. (4.16). The exergy flow of the cycles to the cooling fluid (water) at the



condenser inlet (point e) to the outlet (point f) (Figures 4.4 (a) - (d)) is equally expressed as:

$$\Delta \dot{E}x_{wat} = \dot{m}_{wat}(\dot{e}_f - \dot{e}_e) = \dot{m}_{wat}[(h_f - h_e) - T_o(s_f - s_e)] = 0 \quad (4-17)$$

and the exergy flow rate of working fluid from the vapour–liquid (state 4 or 6) to the cooling liquid (Figures 4.4 (a) - (d)) for the cycles is similarly expressed as:

$$\Delta \dot{E}x_{wf} = \dot{m}_{wf}(\dot{e}_4 - \dot{e}_1) = \dot{m}_{wf}[(h_4 - h_1) - T_o(s_4 - s_1)] = 0 \quad (4-18)$$

Hence, the exergy efficiency of the condensing processes  $\eta_{II,cp}$  of the working fluid were computed to be 75.46%, 75.51%, 75.46% and 75.51% for the simple TFC, TFC with IHE, reheat TFC and regenerative TFC respectively using Eq. (4.19):

$$\eta_{II,cp} = \frac{\Delta \dot{E}x_{wat}}{\Delta \dot{E}x_{wf}} = \frac{\dot{m}_{wat}(\dot{e}_f - \dot{e}_e)}{\dot{m}_{wf}(\dot{e}_4 - \dot{e}_1)} \quad (4-19)$$

### 4.3.3 Exergy Efficiencies of the Cycles

The exergy efficiency  $\eta_{II}$  of a power cycle is the ‘ratio of the actual thermal efficiency  $\eta_{th}$  to the maximum possible (reversible) thermal efficiency  $\eta_{th,rev}$  under same conditions’. That is,

$$\eta_{II} = \frac{\eta_{th}}{\eta_{th,rev}} \quad (4-20)$$

Where  $\eta_{th,rev}$  denotes the reversible thermal efficiency (Carnot equivalent), which can be determined as follows:

$$\eta_{th,rev} = \left(1 - \frac{T_c}{T_h}\right) \quad (4-21)$$

Where  $T_c$  and  $T_h$  denote the temperatures of heat sink (or cold stream) and heat source (or hot streams) respectively. Hence, the exergy efficiency  $\eta_{II}$  of the cycles were computed to be 63.34%, 68.96%, 63.65% and 66.05% for the simple TFC, simple TFC with IHE, reheat TFC and regenerative TFC respectively using Eq. (4.20).

The exergy efficiency measures the performance of actual processes in relation to the performance of the corresponding reversible processes. This enables the comparison of the performance of different devices or systems that are designed to do the same task on the basis of their efficiencies, illustrating the better the design, the lower thermodynamic irreversibilities and the better exergy efficiency.

#### 4.4 Results and Discussion

The performance parameters of the trilateral flash cycles (TFCs) with the *n*-pentane as working fluid were examined thermodynamically, from the resource and technology viewpoints. The process simulations and a detailed thermodynamic analysis of the simple TFC, TFC with internal heat exchanger (IHE), reheat TFC and regenerative TFC, and their ancillary components at the subcritical operating conditions were conducted and results obtained (Appendices A.1 to A.4).

The computed thermodynamic properties for temperature, pressure, enthalpy and entropy of the various cycles' thermodynamic states at the cycle high

temperature of 473 K and average condensing temperature of 309 K is listed in Table 4.1.

**Table 4.1:** Cycle thermodynamic properties

Cycle	State no.	Temperature [K]	Pressure [kPa]	Enthalpy [kg/kJ]	Entropy [kg/kJ-K]
Simple TFC	1	298.2	101.3	23.36	0.07772
	2	298.2	3,000	28.37	0.07772
	3	473	3,000	638.7	1.583
	4	309	101.3	499.6	1.583
TFC with IHE	1	298.2	101.3	23.36	0.07772
	2	298.2	3000	28.37	0.07772
	3	301	3000	465.5	1.144
	4	473	3000	638.7	1.583
	5	371.9	101.3	499.6	1.583
	6	309	101.3	487.8	0.6332
Reheat TFC	1	298.2	101.3	23.36	0.07772
	2	298.2	3000	28.37	0.07772
	3	473	3000	638.7	1.583
	4	371.9	1800	622.8	1.583
	5	433	1800	574.8	1.478
	6	309	101.3	461.7	1.478
Regenerative TFC	1	298.2	101.3	23.36	0.07772
	2	298.2	300	23.71	0.07778
	3	412	300	112.87	0.3504
	4	461.7	3000	117.84	0.3511
	5	473	3000	638.74	1.583
	6	309	101.3	513.52	1.62
	7	412	101.3	583.06	1.796

Tables 4.2 and 4.3 present the comparison of the heating and condensing processes of working fluid in the cycles. It is observed that the exergy efficiencies of the heating processes were 86.15%, 93.46%, 87.05% and 88.74% respectively for the simple TFC, simple TFC with IHE, reheat TFC and

regenerative TFC. Their corresponding exergy efficiencies of the condensing processes are 75.46%, 75.51%, 75.51% and 75.46% respectively.

The exergy efficiencies of the cycles were computed to be 63.34%, 68.96%, 63.65% and 66.05% for the simple TFC, TFC with IHE, reheat TFC and regenerative TFC respectively. Compared with the simple TFC, the exergy efficiency of the TFC with IHE improved by about 8.9% over the simple TFC while those of the regenerative TFC and reheat TFC improved approximately by 4.3% and 0.5% respectively. This is due to the lower exergy destruction rates in the heat exchanger and/ or the condenser.

**Table 4.2:** Thermodynamic properties of the heating process of the working fluid

Simple TFC	Heat source temperature [K]		Point a <sup>1</sup>	473
			Point b <sup>1</sup>	369
	Working fluid temperature [K]		State 2 <sup>1</sup>	298.2
			State 3 <sup>1</sup>	473
<b>Exergy efficiency [%]</b>				<b>86.15</b>
Simple TFC with IHE	Heat source temperature [K]		Points a <sup>1</sup>	473
			Points b <sup>1</sup>	406.9
	Working fluid temperature [K]		State 2 <sup>1</sup>	298.2
			State 4 <sup>1</sup>	473
<b>Exergy efficiency [%]</b>				<b>93.46</b>
Reheat TFC	Heat source temperature [K]		Point a <sup>1</sup>	473
			Point b <sup>1</sup>	368.6
	Working fluid temperature [K]		State 2 <sup>1</sup>	298.2
			State 3 <sup>1</sup>	473
<b>Exergy efficiency [%]</b>				<b>87.05</b>
Regenerative TFC	Heat source temperature [K]		Point a <sup>1</sup>	473
			Point b <sup>1</sup>	406.9
	Working fluid temperature [K]		State 2 <sup>1</sup>	298.2
			State 5 <sup>1</sup>	473
<b>Exergy efficiency [%]</b>				<b>88.74</b>

<sup>1</sup>Refer to points a, and b, and states 2 and 3 respectively in Figure 4.3

**Table 4.3:** Thermodynamic properties of the condensing process of the working fluid

Simple TFC	Working fluid temperature [K]	State 1 <sup>2</sup>	298.2
		State 4 <sup>2</sup>	309
	Cooling fluid temperature [K]	Point e <sup>2</sup>	293
		Point f <sup>2</sup>	297
<b>Exergy efficiency [%]</b>			<b>75.46</b>
TFC with IHE	Working fluid temperature [K]	State 1 <sup>1</sup>	298.2
		State 6 <sup>1</sup>	309
	Cooling fluid temperature [K]	Point e <sup>2</sup>	293
		Point f <sup>2</sup>	297
<b>Exergy efficiency [%]</b>			<b>75.51</b>
Reheat TFC	Working fluid temperature [K]	State 1 <sup>2</sup>	298.2
		State 6 <sup>2</sup>	309
	Cooling fluid temperature [K]	Point e <sup>2</sup>	293
		Point f <sup>2</sup>	297
<b>Exergy efficiency [%]</b>			<b>75.51</b>
Regenerative TFC	Working fluid temperature [K]	State 1 <sup>2</sup>	298.2
		State 6 <sup>2</sup>	309
	Cooling fluid temperature [K]	Point e <sup>2</sup>	293
		Point f <sup>2</sup>	297
<b>Exergy efficiency [%]</b>			<b>75.46</b>

<sup>2</sup>Refer to points e and f, and states 1, 4 and 6 respectively in Figure 4.4.

## 4.5 Conclusions

This study presents the process simulations and detailed thermodynamic analyses of the trilateral flash cycle (TFC) systems as a solution to improve the system efficiency and to create a more secure, sustainable and affordable energy system. The thermodynamic performance of the simple TFC, TFC with internal heat exchanger (IHE), reheat TFC and regenerative TFC operating at the subcritical conditions with low-grade heat in the temperature limit of 393 K – 473 K for heat recovery-to-power generation were examined. Their corresponding heat transfer from the heat source to the power cycles,

thermodynamic efficiency of the power cycles, and heat dissipation from the power cycles to the heat sink.

The exergy efficiencies of heating and condensing processes of the simple TFC were 96.55% (Eq. 12) and 75.46% (Eq. 19) respectively, compared with 93.46% (Eq. 12) and 75.51% (Eq. 19) for the TFC with IHE, 88.74% (Eq. 12) and 75.51% (Eq. 19) for the regenerative TFC, and 87.05% (Eq. 15) and 75.46% (Eq. 19) for the reheat TFC. The exergy efficiencies of the simple TFC was computed to be 63.34%, compared with 68.96% for the TFC with IHE, 66.05% for the regenerative TFC and 63.65% for the reheat TFC (Eq. 20). These suggest that the integration of IHE, reheating and fluid-feed heating in the TFC enhanced the heat exchange performance and as well the system efficiency.

Exergy-based assessment of TFC systems was carried out to permit a realistic modelling, prediction of behaviour and application of TFCs for waste heat recovery-to-power. Moreover, it provides a valuable evaluation of TFCs from strictly thermodynamics perspective.

**CHAPTER FIVE**

**PERFORMANCE OPTIMIZATION AND PARAMETRIC  
STUDY OF TRILATERAL FLASH CYCLES**

## 5 Introduction

In recent times, there is a growing interest in waste heat recovery-to-power technologies because as much as 20 - 50% of the total industrial primary energy consumed or the heat generated is exhausted as waste heat (Hung et al., 1997; Johnson et al., 2008; Lian et al., 2011; Fang et al., 2013) and from numerous other sources; which are generally low-grade (Chen, 2010). As a result of the absence of efficient low-grade waste heat recovery-to-power systems (Franco and Casarosa, 2002; Chen et al., 2011a), low-grade heat has been under-exploited, resulting in severe environmental (thermal) pollution (Dai et al., 2009b).

The mounting pressure for more power generation from alternative sources and improving energy efficiency, basically motivated by the severe environmental challenges have promoted the heat recovery-to-power generation applications. As a result, a number of innovative solutions, which have been developed/proposed for sustainable and affordable power generation, had made low-grade waste heat recovery-to-power generation become feasible.

The trilateral flash cycle (TFC) has been studied for heat recovery-to-power generation (Smith, 1993; Smith and da Silva, 1994; Fischer, 2011). Yet, its application has not been achieved due to concerns about thermodynamic and technical feasibilities, safety as well as the expansion of a two-phase mixture in an expander. The expander, a technically most challenging component, may be a variable phase turbine, a scroll expander, a screw expander or a reciprocating engine (Fischer, 2011). The cycle feasibility depends on the efficient isentropic



expansion of light hydrocarbons from the saturated liquid phase into the two-phase region. This is performed most efficiently with a Lysholm twin-screw expander particularly when the exhausted vapour is wet (Smith and da Silva, 1994). A screw expander has also been proposed as suitable for the TFC application, as it can reach isentropic efficiency of over 75% (Smith et al., 2001). Moreover, expanders available from a good number of original equipment manufacturer, used for LNG applications involving two-phase fluids attain isentropic efficiencies of about 95% (Kaupert et al., 2013).

With the present status of development in the energy conversion technology, large combined-cycle systems are exceeding 70% of the maximum theoretical (Carnot) efficiency. Though, this accomplishment is only achieved with power plants above 300 MW at a large scale. Novel heat recovery-to-power cycles and designs are vital at feasible costs to achieve this similar remarkable performance on a smaller scale (Korobitsyn, 1998).

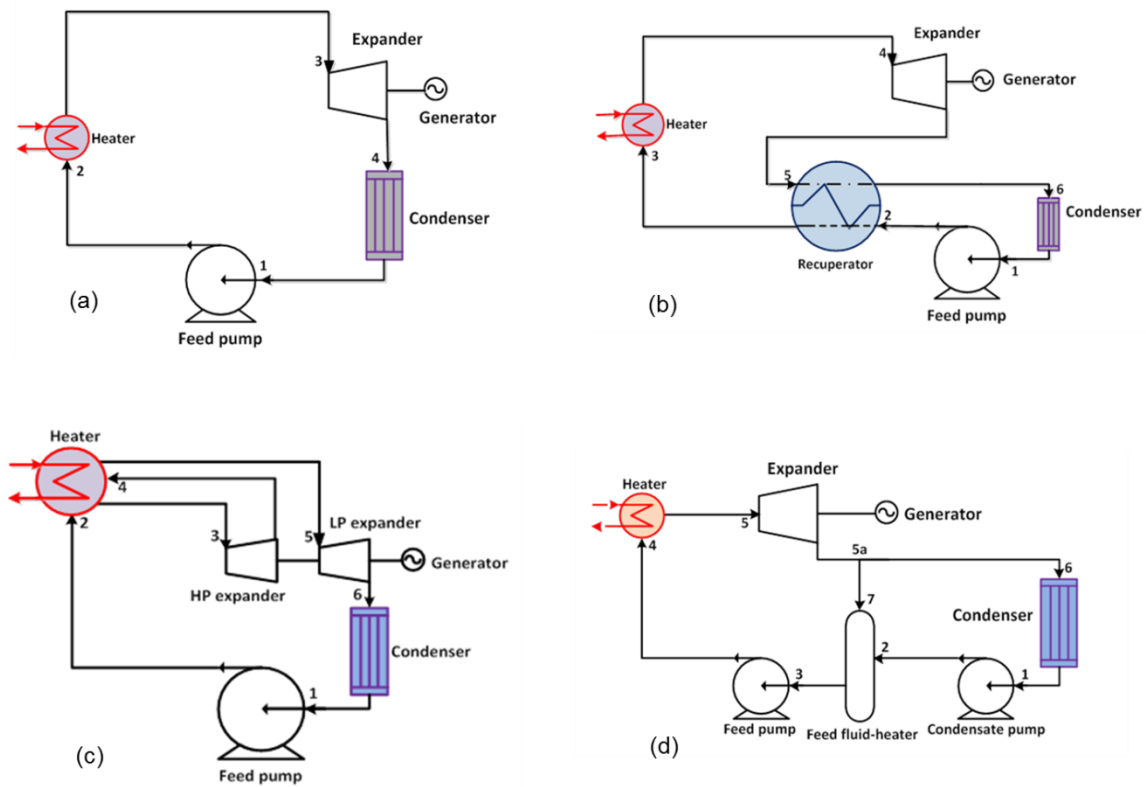
Developments within the existing heat recovery-to-power cycles and an approach towards complex cycles: combinations of basic cycles on the basis of system synthesis (Jin and Ishida, 1993; Jin et al., 1997) are the promising options for improved performance of small- and medium-scale energy conversion technologies. Advanced and combined-cycle systems are reputable due to the fact that no single cycle provides high efficiency as a result of the underlying limitations and the impossibility to function within a wide temperature range. The shortcomings experienced by the single cycles are not suffered by the combined cycles because the bottoming cycle employed the heat rejected by the topping cycle (Korobitsyn, 1998). According to the temperature level,

basic cycles are integrated: high maximum temperature cycles as the topping application while intermediate and/ or low maximum temperature cycles at the bottoming application.

Generally, the application of the TFC systems would be as a bottoming cycle of the gas and/ or steam Rankine cycle to form a combined-cycle system for sustainable and affordable power supply. This study examines the parametric study of performance parameters of the TFC systems for low-grade heat recovery-to-power generation.

## **5.1 System Configurations**

The trilateral flash cycle (TFC) systems for low-grade waste heat recovery-to-power generation from by-product (waste heat) of numerous combustion processes like gas turbine or other sources of heat were examined. Four distinct system configuration designs of the TFC: the simple TFC, TFC with internal heat exchanger (IHE), reheat TFC and TFC with feed fluid-heating (or regenerative TFC) were parametrically analysed.

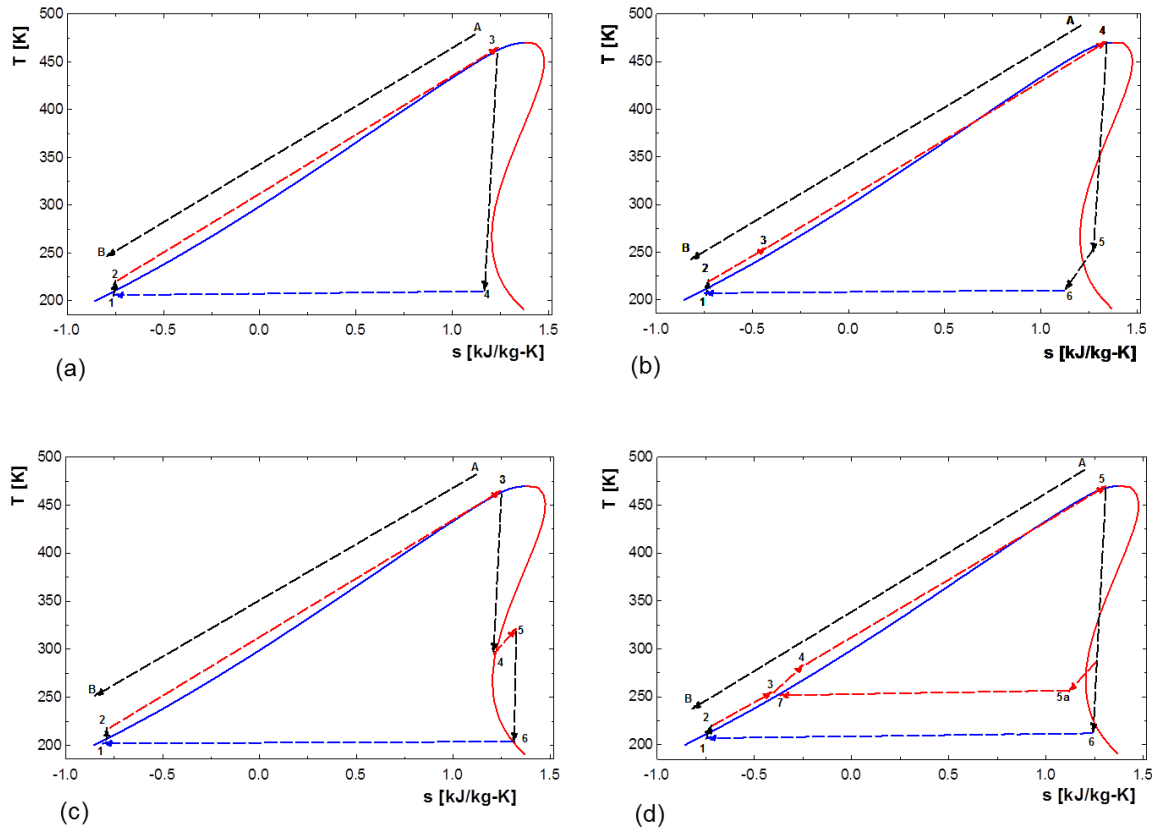


**Figure 5.1:** The trilateral flash cycles, showing the schematic cycle configurations of the simple TFC in (a), TFC with IHE in (b), reheat TFC in (c) and regenerative TFC in (d)

Figures 5.1 and 5.2 depict the schematic cycle configurations and the temperature–entropy (T–s) diagrams of their thermodynamic processes respectively.

Figure 5.1 (a) depicts the simple TFC schematic configuration. Like the Rankine cycle, the simple TFC (Figure 5.2 (a)) comprises four ideal thermodynamic processes. These are the isentropic compression of working fluid in the feed pump (process 1 - 2) during which work is performed on the cycle, constant pressure high-temperature heat addition in the heat exchanger (process 2 - 3), isentropic expansion of high pressurized heated working fluid in the expander (3 - 4) during which work is done by the cycle and constant pressure low-

temperature heat rejection from the working fluid in the condenser (process 4 - 1). Figures 5.1 (b) - (d) depict the schematic cycle configurations of the TFC with IHE, reheat TFC and regenerative TFC respectively. The constant pressure latent heat extraction by the IHE at the expander exhaust (states 3 and 5 in Figure 5.2 (b)) depicts the addition of IHE to the simple TFC. The constant pressure high-temperature heat addition in the heat exchanger (process 4 - 5 in Figure 5.2 (c)) depicts the reheating of reheat TFC, while the constant pressure fluid bleeding to preheat the sub-cooled liquid at the condensate pump outlet (process 5a - 7 in Figure 5.2 (d)) depicts the regeneration process of the regenerative TFC.



**Figure 5.2:** The T–s diagrams of trilateral flash cycles, showing the thermodynamic process of the simple TFC in (a), TFC with IHE in (b), reheat TFC in (c) and regenerative TFC in (d)

### 5.1.1 Working Fluid

*n*-pentane, a dry fluid (Dai et al., 2009b); which its thermo-physical properties are well-suited for low-grade heat recovery-to-power generation within the temperature limits under consideration was adopted as the working fluid for this study. This is because it displays a strong positive slope on the temperature – entropy (T–s) diagram and its saturated liquid expansion tends to dry out at temperatures slightly exceeding 453 K (Smith and da Silva, 1994). Moreover, it possesses good thermo-physical properties (e.g. high critical temperature and pressure), low-cost, and a boiling point slightly above the room temperature.

### 5.1.2 Systems Modelling and Optimization

For the described systems, steady-state steady-flow models of the cycles corresponding to their thermodynamic processes were thermodynamically established and implemented employing EES (engineering equation solver) (Klein, 2013). The models process simulations with model input parameters at subcritical operating conditions were carried out and results obtained.

The process simulation results of the *n*-pentane based TFCs were computed based on the inlet pressure of 3 MPa, expander isentropic efficiency of 90% at the cycle high temperature of 473 K and average condensing temperature of 309 K.

In this present study of the TFCs designed for waste heat recovery-to-power generation; a thermodynamic optimization search technique aimed at finding the operating parameters that optimize net work output was carried out. That also corresponds to optimizing system efficiency only if the mass flow rate of

heat source and its temperature is fixed. The optimization process and the irreversibilities minimization (entropy generation minimization) analysis determine the optimum operating parameters for the cycles, which maximise performance metrics at the cycle high temperature and reduce exergy destruction of the plants.

The following conditions were taken into consideration as a matter of general rule:

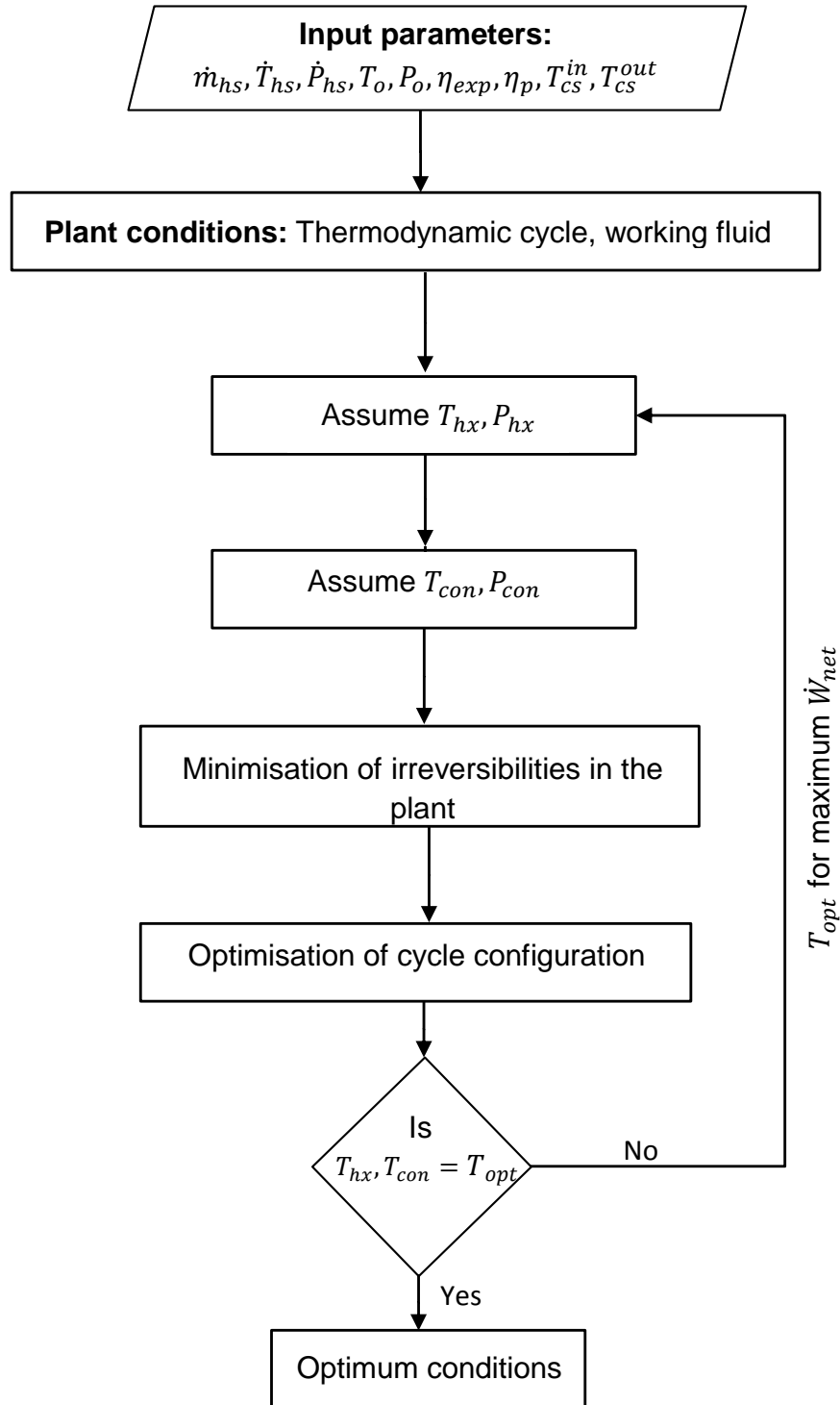
- i. The condensing temperature must be kept at a temperature as low as possible; and
- ii. Optimal heating temperature results of optimization should be kept as high as possible at the overall heat recovery efficiency.

With the analysis of a power cycle and the working fluid combination, the optimal subcritical operating conditions; that is, for heating and condensing temperatures, were determined for cycle net work output per unit mass flow rate of working fluid: the target function of optimization.

Figure 5.3 depicts the simulation algorithm of performance optimization of the cycles. The optimization process was programmed in accordance with the optimization simulation algorithm of the cycles. Using the fixed input parameters at the start of the optimization computation, the simulation algorithm computed the net work output per unit mass flow rate of working fluid for all the cycles.

The results were analysed for all the cycles and their operating characteristics established within the viable design configuration parameters. Thus, the

distinctive operating characteristics for all cycles were evaluated and compared based on the optimisation outcomes.



**Figure 5.3:** The flow chart of simulation algorithm for performance optimization

## 5.2 Parametric Analysis

With the defined values of the model input parameters, standard process simulations of the models were performed at the subcritical operating conditions. A detailed parametric analysis was then carried out by process simulation experimentations with the main models input parameters: the expander inlet pressure at different cycle high temperatures and the expander isentropic efficiency to study the effects of varying these input parameters on system performance of the cycles. That was examined as a function of the expander inlet pressure at cycle high temperature and not temperature because the main constraint in waste heat recovery-to-power technology is the heat source and not the feed pump pressure.

Using the defined model input parameters: the expander inlet pressure of 3 MPa, expander isentropic efficiency of 90% at the cycle high temperature of 473 K and average condensing temperature of 309 K; various cycles' thermodynamic states were computed as well as the performance metrics of the cycles. The parameter sensitivity studies of the models to the expander inlet pressure at the cycle high temperature and expander isentropic efficiency were conducted by process simulation experimentations of pressures over 2 - 3 MPa at the cycle high temperatures of 463 K, 468 K and 473 K respectively and customized expander isentropic efficiency limit of 50 - 100%.

## 5.3 Results and Discussion

A detailed parametric analysis was conducted to examine the effects of key input parameters on performance metrics of the trilateral flash cycles (TFCs) at

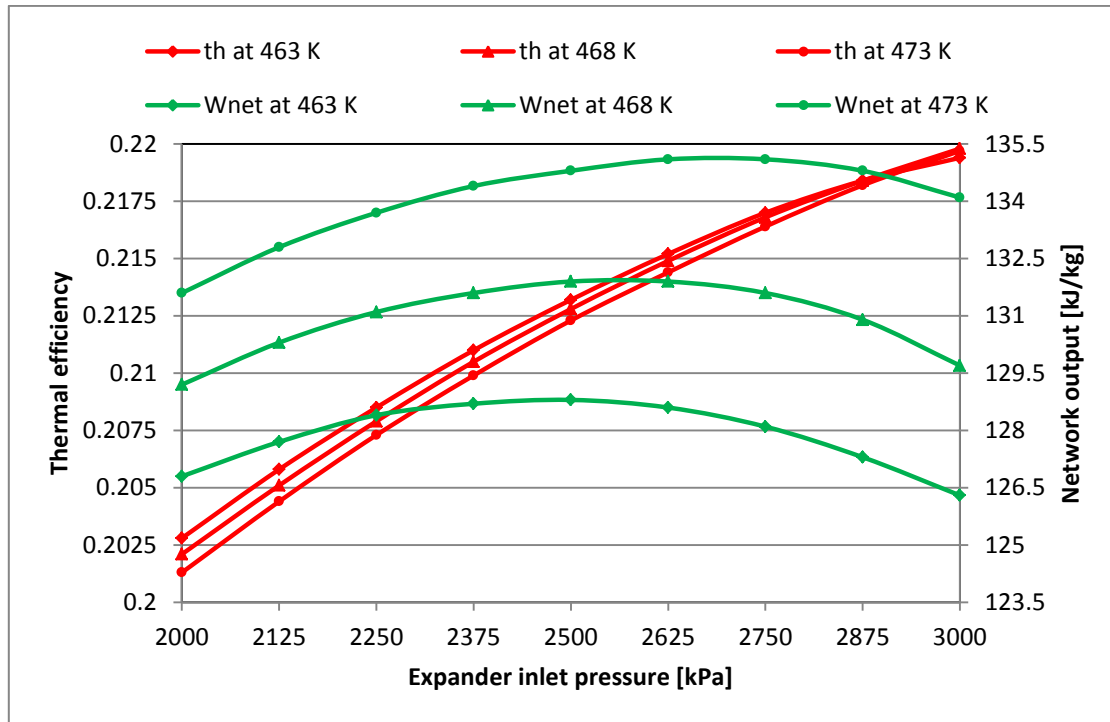


the subcritical operating conditions for heat recovery-to-power generation were examined.

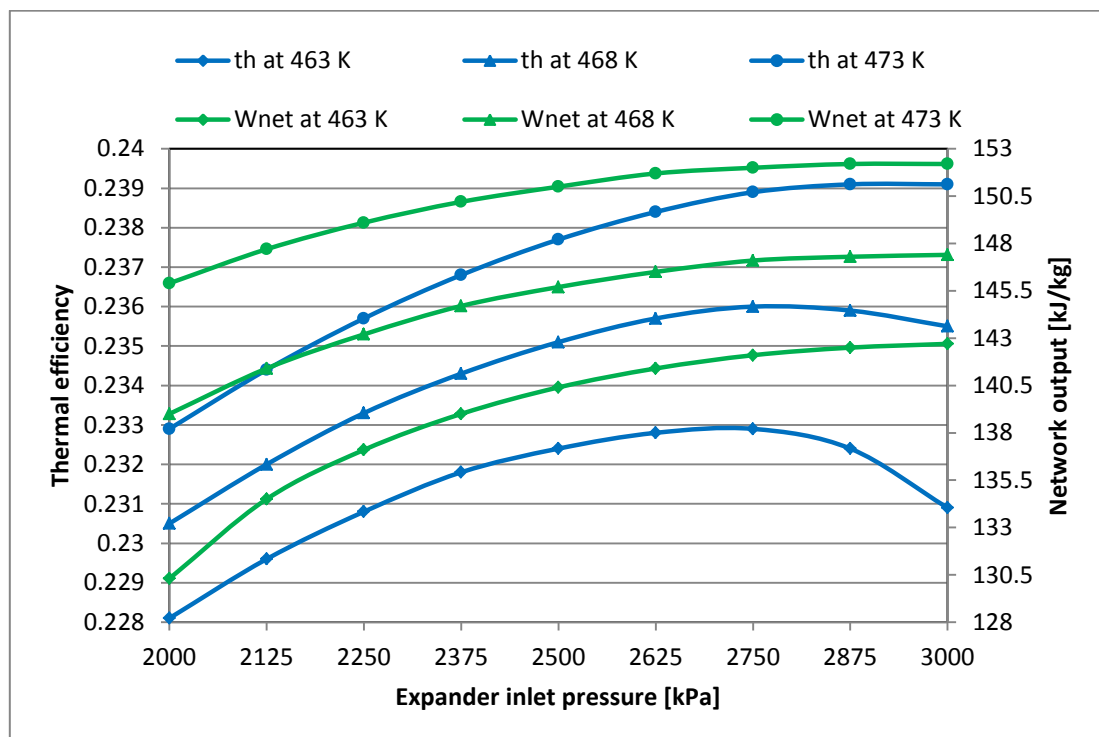
From the process simulation experimentations, the system performance of the cycles and behavioural curves were presented to depict the effects of variation of expander inlet pressures at the cycle high temperature and expander isentropic efficiency parameters on the system performance.

The system configuration designs of the cycles considered were the simple TFC, TFC with internal heat exchange (IHE), reheat TFC and regenerative TFC. The *n*-pentane based cycles were simulated with operating parameters of expander inlet pressure of 3 MPa, expander isentropic efficiencies 90% at the cycle high temperature of 473 K and average condensing temperature of 309 K. The parametric sensitivity studies were performed by varying the expander inlet pressure over 2 - 3 MPa at the cycle high temperatures of 463 K, 468 K and 473 K and expander isentropic efficiency limit of 50 - 100% to study their effects on the performance metrics.

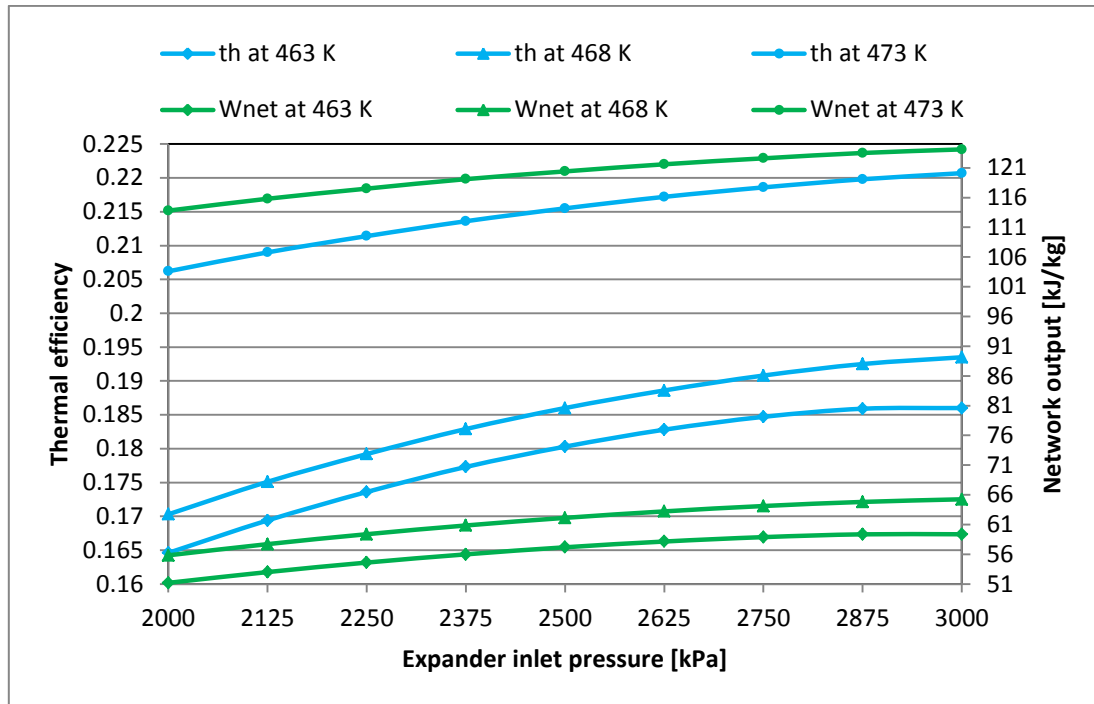
The results of the thermal efficiency and net work output are presented ([Appendices A.1 to A.4](#)). The behavioural plots of performance metrics of the cycles are presented in [Figures 5.4 and 5.5](#).



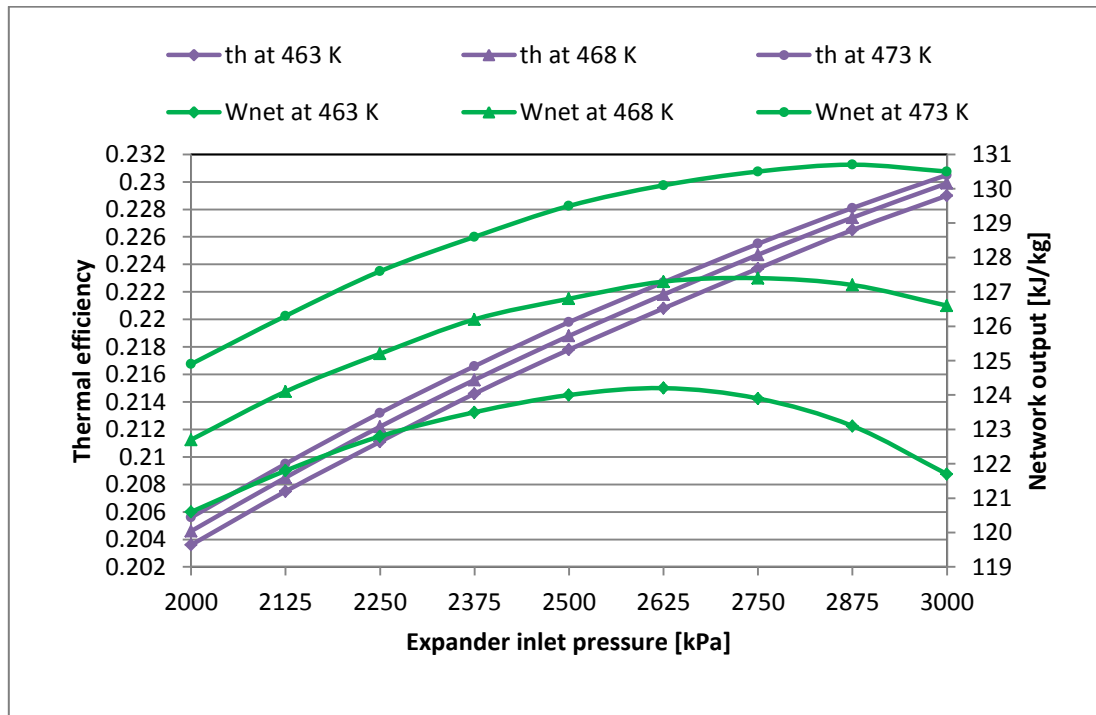
(a)



(b)



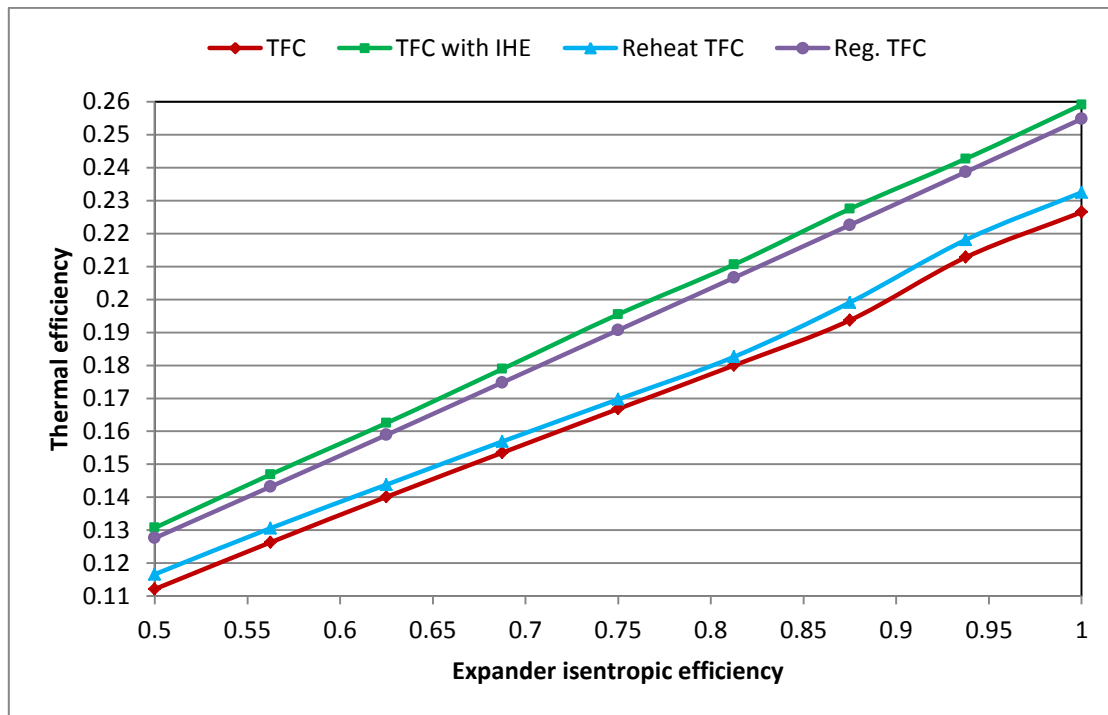
(c)



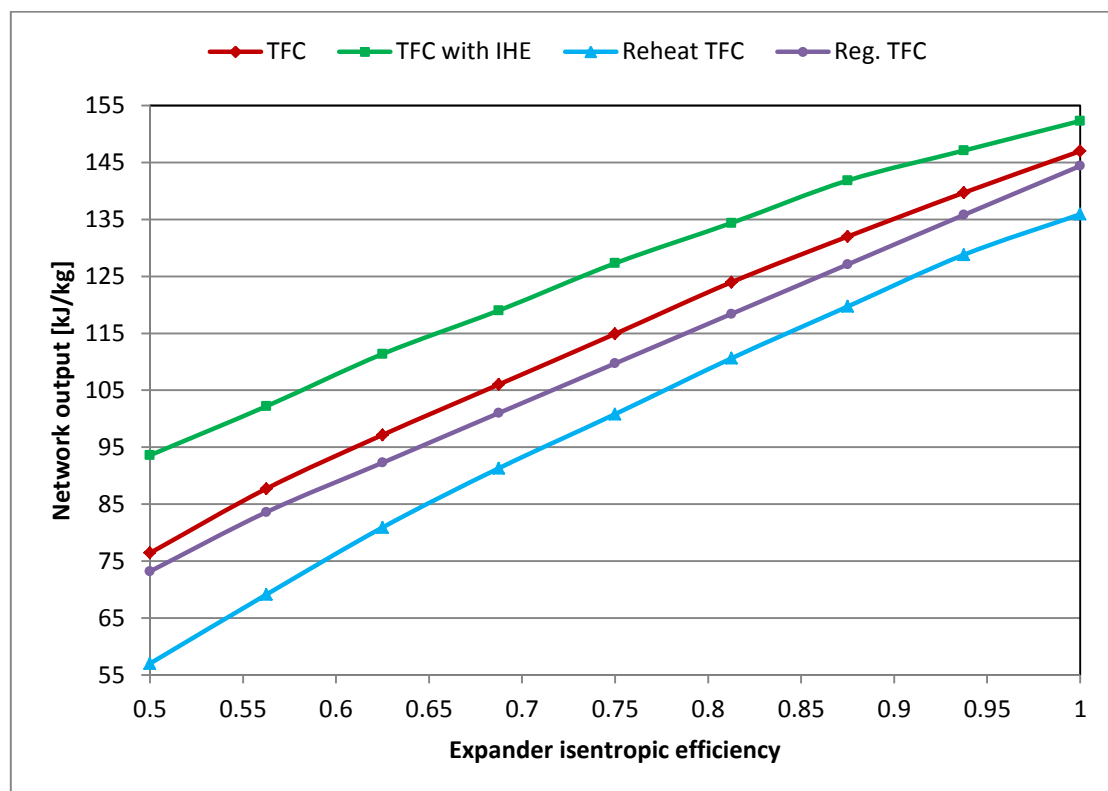
(d)

**Figure 5.4:** The variation of thermal efficiency and net work output with expander inlet pressure at the cycle high temperatures of 463 K, 468 K and 473 K; showing the results of the simple TFC in (a), TFC with IHE in (b), reheat TFC in (c) and regenerative TFC in (d)

The results of the thermal efficiency and net work output variation with expander inlet pressure at the cycle high temperatures of 463 K, 468 K and 473 K is given in Figure 5.4. Figures 5.4 (a) - (d) depict the variation of the thermal efficiencies and net work outputs of the simple TFC, TFC with IHE, reheat TFC and regenerative TFC respectively with expander inlet pressure. It is observed that there were maximal limiting pressures and optimal pressures at each cycle high temperature for the thermal efficiencies and the corresponding net work outputs of the cycles respectively. The resulting thermal efficiencies of the cycles increased as their inlet pressure increased from 2 - 3 MPa. Their corresponding net work outputs increased with increasing inlet pressure until respective optimal pressure were attained and then diminished with increasing pressure. At the cycle high temperature of 463 K for instant, the thermal efficiencies of all the cycles increased from 20.28 - 21.94%, 22.29 - 23.09%, 16.46 - 18.06% and 20.56 - 23.05% for the simple TFC, TFC with IHE, reheat TFC and regenerative TFC respectively. Their corresponding net work outputs increased from 131.6 - 134.1 kW, 145.9 - 152.2 kW, 113.8 - 124.1 kW and 124.9 - 130.5 kW respectively. The increases were due to the addition of maximum quantities of thermal energy per unit mass of the working fluid being injected into the expander, which increase proportionately to the expander inlet pressure; suggesting optimal pressure ratio between the expander inlet and its outlet increases.



(a)



(b)

**Figure 5.5:** Performance variation with expander isentropic efficiency

The results of the thermal efficiency and net work output variation with expander isentropic efficiency are given in Figure 5.5. Figures 5.5 (a) and (b) depict the variation of the thermal efficiencies and net work outputs of the simple TFC, TFC with IHE, reheat TFC and regenerative TFC respectively with expander isentropic efficiency. The resulting thermal efficiencies of the cycles increased proportionately as the expander isentropic efficiency values increased. Their corresponding net work outputs increase almost proportionately with increasing expander isentropic efficiency values. The thermal efficiencies of all the cycles increase from 11.21 - 22.65%, 13.07 - 25.91%, 11.66 - 23.25% and 12.76 - 25.48% for the simple TFC, TFC with internal heat exchange (IHE), reheat TFC and regenerative TFC respectively. Their corresponding net work outputs increase from 76.41 - 147 kW, 93.58 - 152.3 kW, 57 - 135.9 kW and 73.18 - 144.4 kW respectively; when the expander isentropic efficiency values were increased from 50 - 100%. The increases in the thermal efficiencies were due to the significant increase in the expander work outputs, which suggest optimal pressure ratios between the expander inlets and their outlets increased as the isentropic efficiencies increase.

## 5.4 Conclusions

The parametric analyses of the trilateral flash cycle (TFC) systems using *n*-pentane as working fluid for low-grade waste heat recovery-to-power generation were presented as a solution to evaluate the effects of key thermodynamic parameters on the performance metrics. Their predictions can be used for guiding system design, improve understanding of the cycles' operation and behaviour of performance metrics. The parametric study of the simple TFC,

TFC with internal heat exchanger (IHE), reheat TFC and regenerative TFC operating at the subcritical conditions with low-grade heat of 393 K - 473 K were examined.

The models process simulation results of the *n*-pentane based TFCs were computed based on the inlet pressure of 3 MPa, expander isentropic efficiency of 90% at the cycle high temperature of 473 K and average condensing temperature of 309 K. Parametric sensitivity studies of the cycles were carried out by varying expander inlet pressure at different cycle high temperatures and the expander isentropic efficiency; and their performance metrics based on these input parameters were computed.

Results depict that the thermal efficiencies and net work outputs of all the cycles increase as the inlet pressure at the cycle high temperature increased from 2 - 3 MPa and increase as the expander isentropic efficiency increased from 50 - 100%. The effects of varying expander inlet pressure at cycle high temperature and expander isentropic efficiencies on performance metrics of the cycles were significant. As the expander inlet pressure at the cycle high temperature increased, optimum pressure ratios between the expander inlets and their outlets increased, coupled with an increase in the quantities of thermal energy addition per unit mass flow rate of the working fluid. This resulted in higher efficiencies and increased work outputs. As the expander isentropic efficiencies increased, there are increases in the thermal efficiencies because the expander work outputs increased substantially, which suggest optimal values of pressure ratios.

**CHAPTER SIX**

**OPTIMIZED PERFORMANCE ANALYSIS AND  
COMPARISON OF THE TRILATERAL FLASH CYCLES**



## 6 Introduction

There have been significant quantities of low-grade waste heat dissipated into the environment due to the exhaust waste heat from conventional fossil fuel energy systems and industrial waste heat (Hung et al., 1997; Johnson et al., 2008; Lian et al., 2011; Fang et al., 2013). This has resulted in severe environmental pollution (Dai et al., 2009b). There are numerous renewable sources of energy like geothermal, solar thermal and biomass (Yari, 2009; Fischer, 2011), providing low-to-medium temperature (low-grade) heat sources that are ordinarily under-exploited.

Thus, there is a renewed significance for low-grade heat recovery-to-power generation, motivated by the quest to improve system efficiency, significantly reduced the primary energy demands and emissions (Patil et al., 2009; Fang et al., 2013), and reduce the carbon footprint of power generation (Ziviani et al., 2014). Numerous heat recovery-to-power technologies have been proposed to exploit low-grade heat sources, which are otherwise challenging to convert using conventional Rankine systems (Ziviani et al., 2014).

Unlike the Rankine cycles, the TFC does not either use a boiler or evaporate the working fluid, instead expands its saturated liquid by flash expansion in an expander for heat recovery-to-power generation (Zamfirescu and Dincer, 2008). Due to bypass of isothermal boiling phase, there is a heat transfer without a pinch-point limitation, causing thermal marching between the exergy temperature profiles of the heat source and the working fluids for the merit of

improved system performance (Winterbone, 1997; Zamfirescu and Dincer, 2008).

Four distinct cycle configuration designs of the TFCs: simple TFC, TFC with IHE, reheat TFC and regenerative TFC; have been modelled, implemented and analysed thermodynamically and parametrically. The combined results of the optimized performances of the *n*-pentane based TFCs obtained are discussed in a wider perspective.

## **6.1 Performance Analysis and Comparison of the Cycles**

The optimized performance analyses and comparison of the trilateral flash cycles (TFCs) were performed using an expander inlet pressure at the cycle high temperature and average condensing temperature. The optimization results of cycles were computed using an expander inlet pressure of 3 MPa and expander isentropic efficiency of 90% at the cycle high temperature of 473 K and average condensing temperature of 309 K. The optimized operating parameters sensitivity analysis was conducted using an expander inlet pressure of 2 - 3 MPa at the cycle high temperature of 473 K and expander isentropic efficiency limit of 50 - 100%.

### **6.1.1 Thermodynamic Efficiencies and Net work Outputs of the Cycles**

A comparison of the performance metrics of the cycles is given in Figure 6.1. Figures 6.1 (a) - (c) depict the thermal efficiencies, exergy efficiencies and net work outputs of the various cycles at an expander inlet pressure of 3 MPa at the cycle high temperature of 473 K.

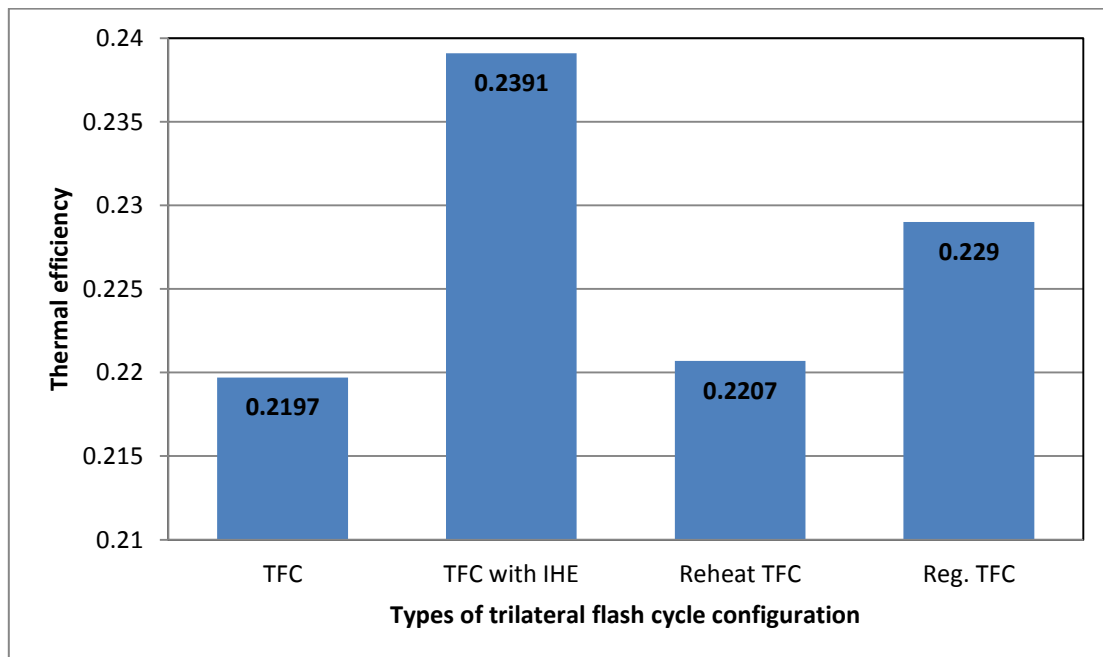
Figure 6.1 (a) depicts that the simple TFC has an optimized thermal efficiency of 21.97%, which is the lowest among the cycles; while that of the TFC with IHE is 23.91%, which is the highest. Compared with the simple TFC, the thermal efficiency of the TFC with IHE has an improvement of about 8.9% over the simple TFC. The regenerative TFC has an optimized thermal efficiency of 22.9%, which is higher than that of the reheat TFC and about a 4.2% improvement over the simple TFC. The reheat TFC has an optimized thermal efficiency of 22.07%, which is a marginal 0.5% improvement over the simple TFC.

Figure 6.1 (b) depicts that the simple TFC at the optimized operating condition has an exergy efficiency of 63.34%, which is the lowest among the cycles; while that of the TFC with IHE is 68.96%, which is the highest. Compared with the simple TFC, the exergy efficiency of the TFC with IHE has an improvement of about 8.9% over the simple TFC due to lower irreversibility rates in the heat exchanger and the condenser. The regenerative TFC has an optimized exergy efficiency of 66.05%, which is higher than that of the reheat TFC and an improvement of about 4.3% over the simple TFC. This is because there is no heat rejection (i.e. lower irreversibility rates) from about 40% of the working fluid in the condenser. The reheat TFC has an optimized exergy efficiency of 63.65%, which is a marginal 0.5% improvement over the simple TFC.

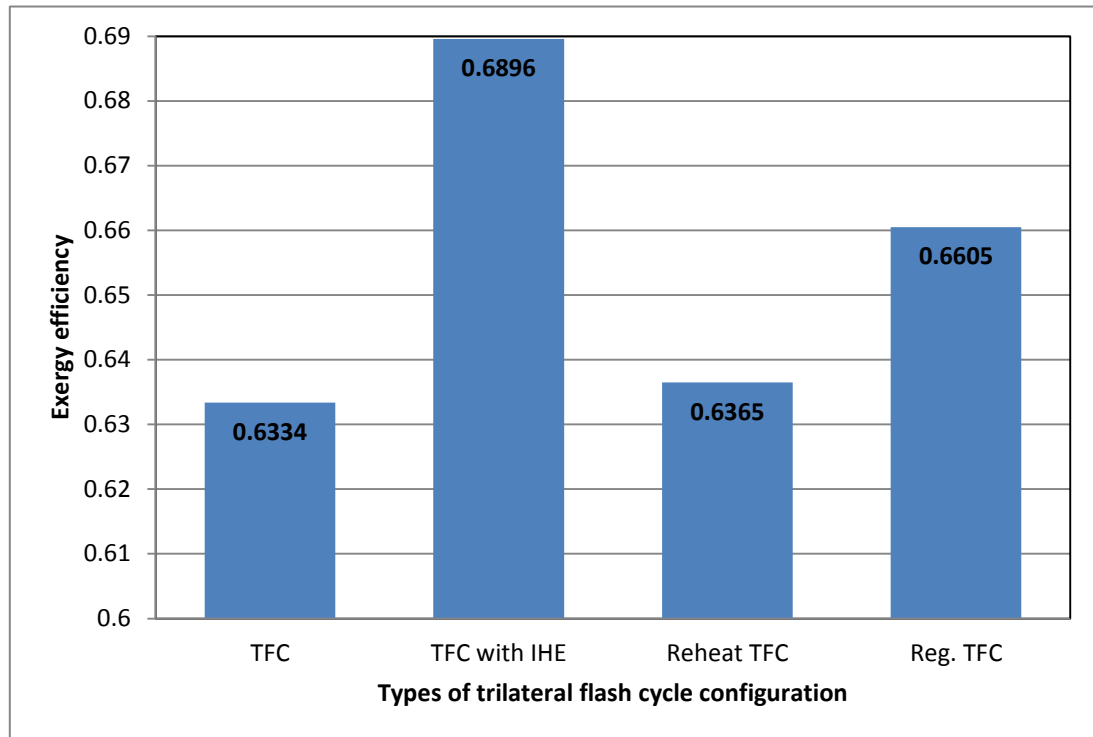
Figure 6.1 (c) depicts that the reheat TFC has an optimized net work output of 124.1 kW, which is the lowest among the cycles; while that of the TFC with IHE is 152.2 kW, which is the highest. Compared with the simple TFC, the net work output of the TFC with IHE shows an improvement of about 13.5% over the

simple TFC. The regenerative TFC has an optimized net work output of 130.5 kW, which is marginally higher than that of the reheat TFC and about 2.7% lower than the simple TFC. Compared with the simple TFC, the net work output of the reheat TFC is about 7.5% lower than the simple TFC.

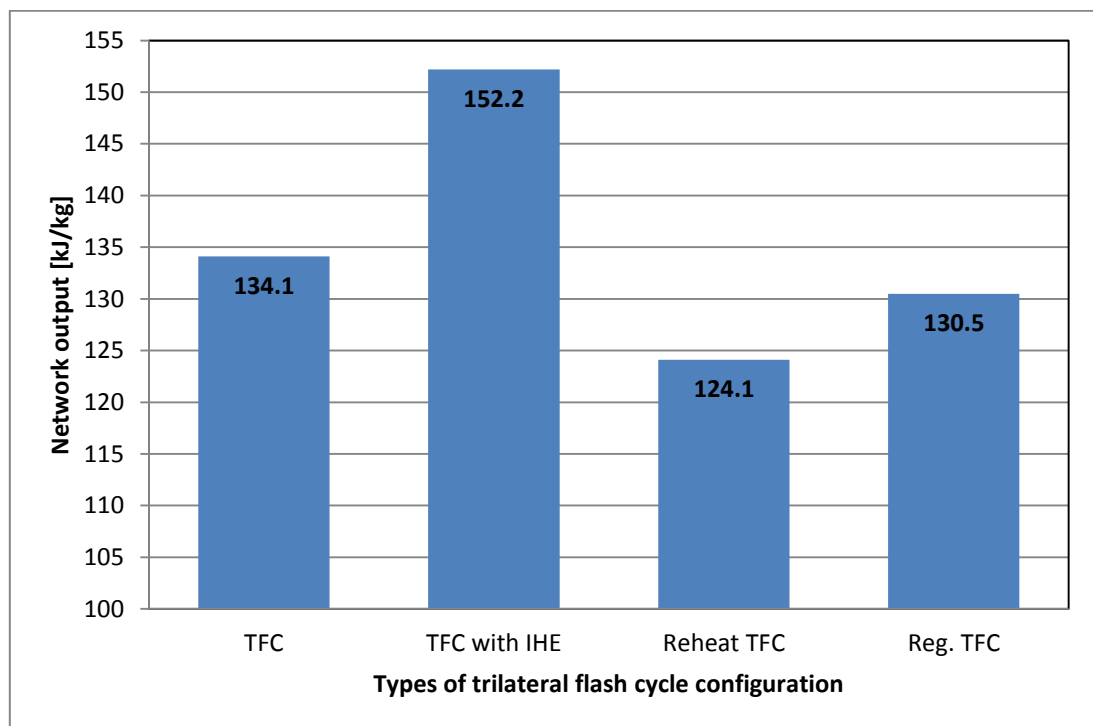
The causes of the differences among the thermal efficiencies and the net work outputs of the cycles are suggested as follow. The IHE extracts and transfers huge quantities of latent heat from the superheated vapour–liquid to preheat the sub-cooled liquid at the feed pump outlet. It lowers the needed heat input, and thus, improves the thermal efficiency of the cycle. Consequently, the quantity of thermal energy addition required to power the TFC with IHE is less than the simple TFC. Moreover, the work output of the cycle is marginally higher than the simple TFC. Thus, the TFC with IHE has a substantially better thermal efficiency than the simple TFC.



(a)



(b)



(c)

**Figure 6.1:** Performance comparison of various trilateral flash cycles

The quantity of thermal energy addition from the reheat TFC and its low pressure expander work output increase when the working fluid temperature increased during the reheating. Though, the magnitude of the increased quantity of thermal energy addition is much higher compared to the net work output. Consequently, the system thermal efficiency is not significantly improved with an increasing quantity of thermal energy addition during reheating.

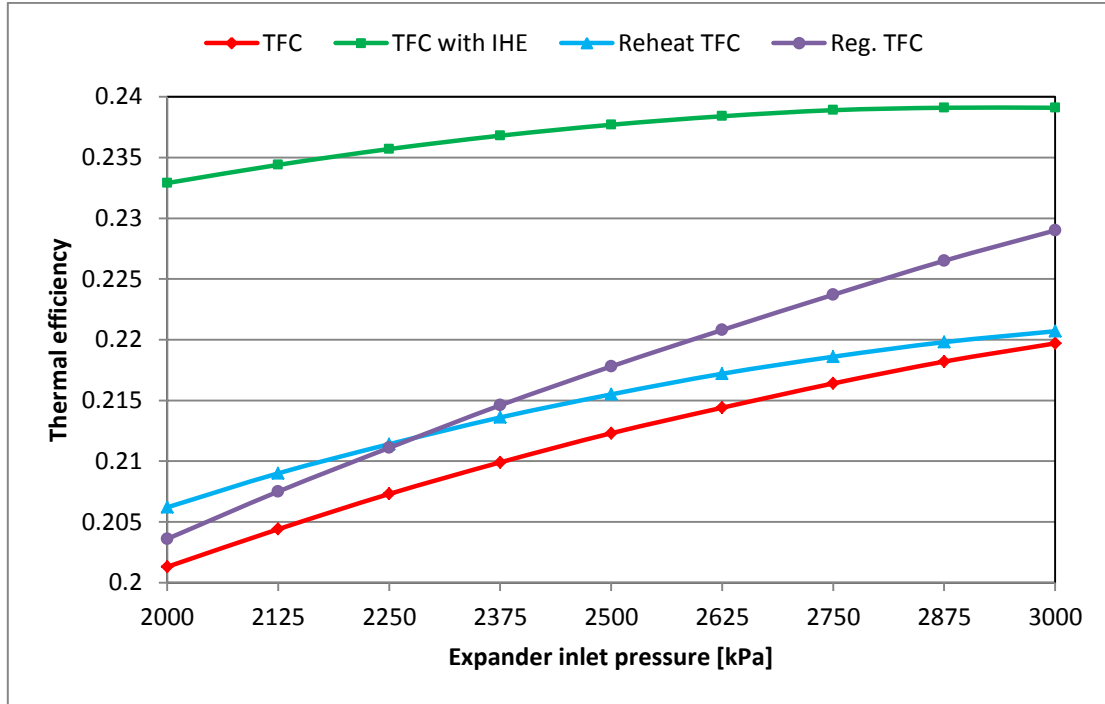
The quantity of thermal energy addition of the regenerative TFC is significantly lower than that of the simple TFC. Yet, the net work output of the cycle decreases because about 40% of the working fluid is bled during the expansion process. Therefore, the thermal efficiency is marginally higher than the simple TFC but lower than that of the TFC with IHE.

### **6.1.2 Sensitivity Studies**

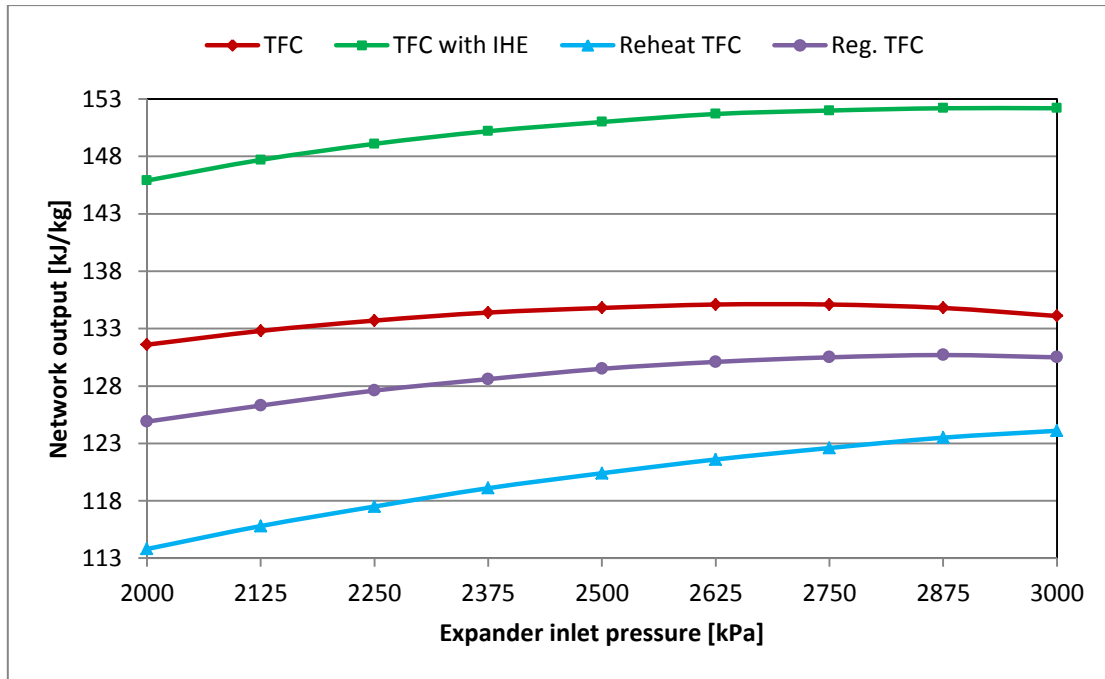
The optimized performances of the cycles have been analysed and compared at the same subcritical operating conditions of expander inlet pressure and expander isentropic efficiency at the cycle high temperature and average condensing temperature.

The results of performance variation with expander inlet pressure at the cycle high temperature of 473 K are given in Figure 6.2. Figures 6.2 (a) and (b) depict the variation in thermal efficiencies and net work outputs of the cycles with expander inlet pressure at the cycle high temperature of 473 K. The results depicts that the resulting thermal efficiencies of the cycles increased as inlet pressure increased from 2 - 3 MPa at the cycle high temperature of 473 K. Their corresponding net work outputs increased with increasing inlet pressure until

respective optimal pressure were attained and then diminish with increasing pressure. The thermal efficiencies of all the cycles increase from 20.13 - 21.97%, 23.29 - 23.91%, 20.62 - 22.07% and 20.36 - 22.9% for the simple TFC, TFC with IHE, reheat TFC and regenerative TFC respectively. Their corresponding net work outputs increased from 131.6 - 134.1 kW, 145.9 - 152.2 kW, 113.8 - 124.1 kW and 124.9 - 130.5 kW respectively. The increases were as a result of the addition of maximum quantities of thermal energy to the cycles that increased proportionately to the expander inlet pressure. While the net work output of the cycle increased marginally. Thus, the thermal efficiencies of all cycles increased. Nevertheless, the TFC with IHE still had the highest thermal efficiency with varying pressure at the cycle high temperature among the cycles. The thermal efficiency of the regenerative TFC is likewise higher than that of the reheat TFC and the simple TFC. The thermal efficiency of the reheat TFC is almost equal to the simple TFC. This suggests that simply reheating the working fluid cannot significantly improve performance.



(a)

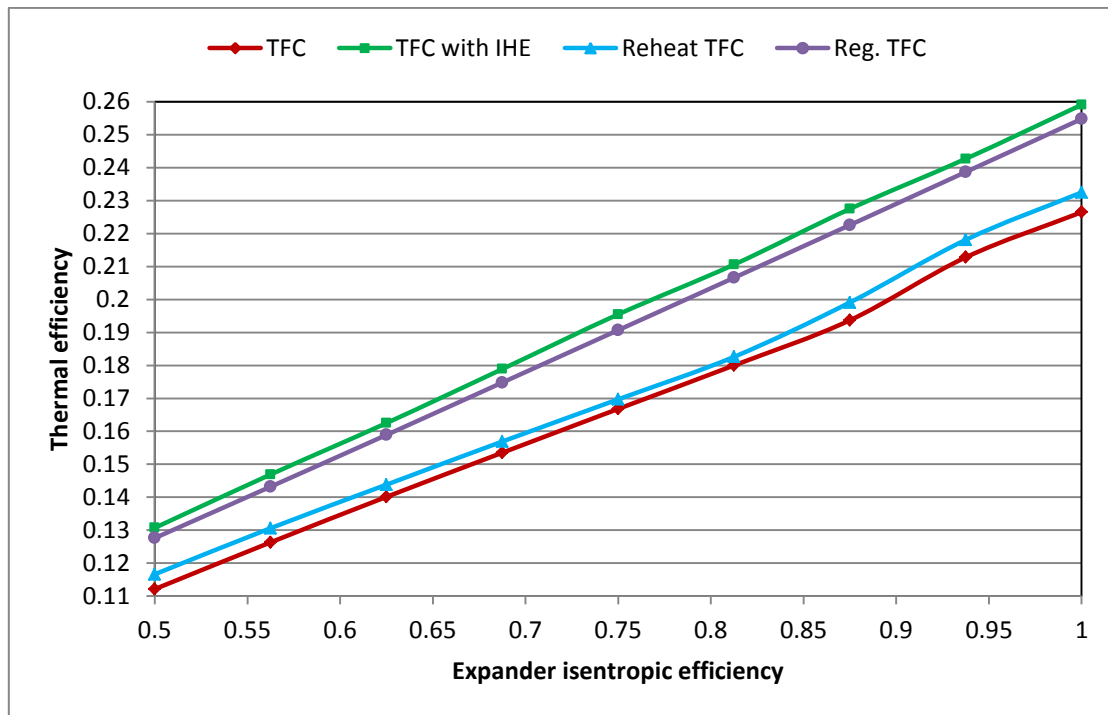


(b)

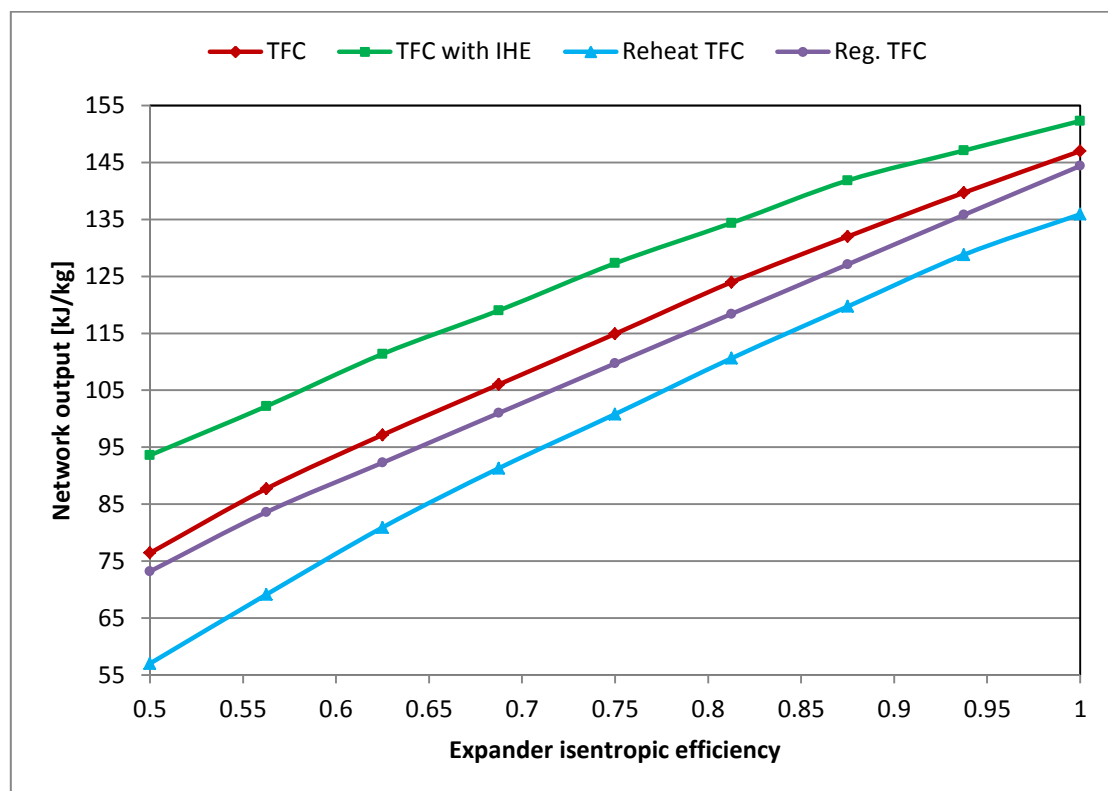
**Figure 6.2:** Performance variation with expander inlet pressure at the cycle high temperature of 473 K



The results of the performance variation with the expander isentropic efficiency are given in Figure 6.3. Figures 6.3 (a) and (b) depict the variation of thermal efficiencies and net work outputs of the cycles with expander isentropic efficiency. The resulting thermal efficiencies of the cycles increased proportionately and their corresponding net work outputs almost proportionately as the expander isentropic efficiency values increased. The thermal efficiencies of all the cycles increased from 11.21 - 22.65%, 13.07 - 25.91%, 11.66 - 23.25% and 12.76 - 25.48% for the simple TFC, TFC with internal heat exchange (IHE), reheat TFC and regenerative TFC respectively. Their corresponding net work outputs increased from 76.41 - 147 kW, 93.58 - 152.3 kW, 57 - 135.9 kW and 73.18 - 144.4 kW respectively when the expander isentropic efficiency values increased from 50 - 100%. The increases in the thermal efficiencies were due to the significant increase in the expander work outputs as the isentropic efficiencies increased. Thus, the thermal efficiencies of all the cycles increased. Nevertheless, the TFC with IHE still had the highest thermal efficiency with varying expander isentropic efficiency among the cycles. The thermal efficiency of the regenerative TFC was also higher than that of the reheat TFC and the simple TFC. The thermal efficiency of the reheat TFC was almost same as the simple TFC.



(a)



(b)

**Figure 6.3:** Performance variation with expander isentropic efficiency

## 6.2 Conclusions

The optimized performance analysis and comparison of the trilateral flash cycles (TFCs) for low-grade waste heat recovery-to-power generation were conducted using an optimization search technique to compute the optimal net work output that also corresponds to optimizing system efficiency at fixed mass flow rate of heat source and its temperature.

Results depicted that the variation in expander inlet pressure at the cycle high temperature and expander isentropic efficiency on the absolute levels attained by performance metrics of the cycles were significant. The thermal efficiencies and net work outputs at the cycle high temperature for all the cycles increased as the inlet pressure increased from 2 - 3 MPa. This is because optimal pressure ratios between the expander inlets and their outlets increased, coupled with an increase in the quantities of thermal energy addition, resulting in higher system performance. In addition, the thermal efficiencies and net work outputs for all the cycles increased as the expander isentropic efficiency increased from 50 - 100%. This is because there were increases in the thermal efficiencies due to the significant increase in the expander work outputs.

## **CHAPTER SEVEN**

### **SUMMARY, CONCLUSIONS AND RECOMMENDATIONS**

## 7 Summary

In the present study, one of the most suitable and efficient ways from proof-of-concept towards a market-ready trilateral flash cycle (TFC) technology for low-grade waste heat recovery-to-power generation: modelling, process optimization and simulation were explored. Modelling and process simulation are valuable tools in the optimization of engineering system design, its performance predictions, development, operation and maintenance. Moreover, innovative thinking about the system configuration design and process that were needed to ensure technology integration and drive towards a step change in investment and commercialization were examined.

The objective of this study is to establish and implement the simulation tools of the TFC for improving the understanding of its operation, physics of performance and to evaluate innovative system configurations. In order to achieve this objective, a rigorous performance study of the TFC and the effects of integration of TFC with an internal heat exchanger (IHE), reheating and regeneration on system performances were examined for low-grade waste heat recovery-to-power generation. Thus, the simple TFC and process integration of TFC with an IHE, reheating and feed fluid-heating to improve the heat exchange performance and system efficiency were investigated.

Furthermore, the screening of promising working fluids was conducted coupled with the choice of the subcritical operating conditions at the inlets of the expander and condenser. Afterwards, steady-state steady-flow models of the TFCs power plants, corresponding to their thermodynamic processes were

modelled and implemented. The thermodynamic properties for temperature, pressure, enthalpy and entropy of all cycles' thermodynamic states at the cycle high temperature of 473 K and average condensing temperature of 309 K were calculated. The results of the models process simulations of the cycles were computed with an expander inlet pressure of 3 MPa and expander isentropic efficiency of 90% at the cycle high temperature of 473 K and average condensing temperature of 309 K. The thermodynamic analyses for all the cycles were evaluated at the cycle high temperature of 473 K and average condensing temperature of 309 K. While the operating parameter sensitivity analyses of the models to the expander inlet pressure at the cycle high temperature and expander isentropic efficiency were examined by the process simulation experimentations of the pressure over 2 - 3 MPa at cycle high temperatures of 463 K, 468 K and 473 K respectively and customized expander isentropic efficiency limit of 50 - 100%.

An in-depth literature review was conducted to provide an introductory investigation of the topic before embarking on a formal research project with the aim to substantiate why this study is significant. Thus, a review of the variety of renewable energies and exhaust waste low-grade heat sources, review of the major power cycles for low-grade heat recovery, working fluids and their selection criteria, expanders for low-grade heat recovery-to-power generation and the energy related terms were highlighted as well as a systemic review of the available studies on waste heat recovery-to-power technologies.

A comprehensive screening of the promising working fluids of the TFCs for heat recovery-to-power generation was conducted and a shortlist of 24 potential

fluids was evaluated. These shortlisted fluids were selected based on meeting the environmental criteria and possessing a positive slope of vapour saturation curve on the temperature–entropy diagram. Of the shortlist, the most promising working fluid, which satisfies the thermo-physical, environmental and techno-economic concerns suitable for the maximum cycle temperature limit of 393 - 473 K at subcritical operating conditions, was selected.

The methodology was designed to systematically solve the research problems and to achieve the set objectives. Four distinct cycle configuration designs of the TFC: the simple TFC, TFC with internal heat exchanger (IHE), reheat TFC and regenerative TFC were examined. The steady-state steady-flow models of the TFCs power plants were thermodynamically modelled and implemented to represent the procedure to system development and the understanding needed to support such process. These embody valuable tools to “Run Simulations” for scenarios that could be apparently unmanageable in the physical experiment, in order to generate relevant information and data.

The thermodynamic solutions of the cycles for low-grade heat recovery-to-power generation were examined at the subcritical operating conditions to assess the performance of the proposed cycles and their predictions. Parameter sensitivity studies of the cycles were performed to investigate the effects of the configuration parameters on the thermodynamic performance of the cycles for waste heat recovery-to-power generation. Parametric optimization of the cycles was carried out using an optimization search technique to find the optimum net work output at the cycle high temperature that corresponds to the optimum system efficiency of the cycle. The resulting optimized performances

were compared and analysed over the same temperature limits at the subcritical operating conditions.

## **7.1 Conclusions**

This study established and implemented simulation tools of four distinct system configuration designs of the TFCs employing *n*-pentane as their working fluid and their optimized performance metrics were examined for low-grade waste heat recovery-to-power generation. The effects of the key operating parameters (variables) on the thermodynamic performance metrics of the cycle were investigated.

Steady-state steady-flow models of different components of the cycles, corresponding to their thermodynamic processes were established and implemented. The models of the cycles were simulated to obtain a good understanding of their behaviour at subcritical operating conditions. The models were afterwards utilized to optimize the cycles, from basic cases and improving them in a realistic procedure to assess the system performance metrics.

By establishing and implementing the simulation models, and examining the effects of the key operating parameters, an improved understanding of the cycles' operation and physics of their performance metrics have been obtained.

The main conclusions drawn from the research outcomes are summarised as follows:



- i. The optimized thermal efficiency of the simple TFC is the lowest among the cycles. The TFC with IHE has the highest, which is higher than that of the regenerative TFC and the regenerative TFC is more than that of the reheat TFC. With operating parameters of an expander inlet pressure of 3 MPa and expander isentropic efficiencies 90% at the cycle high temperature of 473 K and average condensing temperature of 309 K, the computed thermal efficiency of the simple TFC is 21.97%. Compared with the simple TFC, the thermal efficiency of the TFC with IHE has an improvement of about 8.9% over the simple TFC, and the regenerative TFC and the reheat TFC has an improvement of about 4.2% and 0.5% respectively over the simple TFC.
- ii. The exergy efficiency of the simple TFC at the optimized operating condition is the lowest among the cycles. The TFC with IHE is the highest, which is higher than that of the regenerative TFC and the regenerative TFC is higher than that of the reheat TFC. With operating parameters of an expander inlet pressure of 3 MPa and expander isentropic efficiencies 90% at the cycle high temperature of 473 K and average condensing temperature of 309 K, the computed thermal efficiency of the simple TFC is 63.34%. Compared with the simple TFC, the thermal efficiency of the TFC with IHE has an improvement of about 8.9% over the simple TFC, and the regenerative TFC and the reheat TFC has an improvement of about 4.3% and 0.5% respectively over the simple TFC.
- iii. The optimized net work output of the reheat TFC is the lowest among the cycles. The TFC with IHE is the highest, which is more than that of the

simple TFC and the simple TFC is more than that of the regenerative TFC. With operating parameters of an expander inlet pressure of 3 MPa and expander isentropic efficiencies 90% at the cycle high temperature of 473 K and average condensing temperature of 309 K, the computed net work output of the simple TFC is 134.1 kW. Compared with the simple TFC, the net work output of the TFC with IHE has an improvement of about 13.5% over the simple TFC, and those of regenerative TFC and reheat TFC were about 2.7% and 7.5% lower than that of the simple TFC respectively.

- iv. The thermal efficiencies and net work outputs of all the cycles observably increased as the inlet pressure increased from 2 - 3 MPa and increased as the expander isentropic efficiencies increased from 50 - 100%, while their exergy efficiencies increased.
- v. From a comprehensive evaluation of the optimized thermodynamic performances of the cycles, the TFC with IHE attained the best comparative performance metrics among the cycles over same subcritical operating conditions. It is followed by the regenerative TFC whereas the simple TFC and the reheat TFC is the last choice. Though, the goal of evaluating the proposed cycles was not to claim the absolute quality of been at a comparative advantage. But to showcase their distinct features, establish a system configuration design basis, and to understand the interrelationship among key input parameters and system performance metrics.

Nevertheless, the development of a dependable and efficient TFC engines is reputable because they would be the key sub-engines to attain combined-cycle

or complex systems with high thermodynamic efficiency for the grid as well as for distributed power generation. Furthermore, because there is no single cycle that can provide high efficiency as a result of the underlying cycle limitation, which is partially explained by the low-temperature heat of the energy source.

## **7.2 Recommendations for Further Research**

The research outcomes of this study demonstrate that there is a great potential for the application of the proposed TFCs for low-grade waste heat recovery-to-power generation. The following are recommended for future work to further progress substantially and optimise the TFC technology for low-grade heat recovery-to-power generation.

In CHAPTER THREE, a synopsis of the promising working fluids for the TFCs and a shortlist of 24 potential fluids were evaluated. Though, it is comprehensive, a systemic analysis of the behaviour of the different categories of working fluid should be conducted to outline a selection procedure to decide the most promising fluid. That should be able to satisfy the thermo-physical, environmental (ozone depletion and global warming potentials) and techno-economic (toxicity, fluid stability, availability and cost) concerns. Also, zeotropic mixtures of working fluid have been innovatively suggested in order to improve the performance of heat recovery-to-power cycles. The TFCs using pure working fluids and zeotropic fluid mixtures should be studied to optimize heat recovery-to-power generation from renewable energies and waste heat. Furthermore, a selection procedure on the suitable choice of working fluids for various heat sources and heat sink profiles should be established for the

proposed cycles. This is required in order to adapt and take full advantage of the cycle performances for renewable energies and waste heat recovery-to-power generation.

This study does not include the improvement scheme of the TFC with a combination of the feed fluid-heating (regenerator) and the internal heat exchanger (recuperator) examined in CHAPTER THREE. This can also be employed to optimize the design configuration of the cycle for improved heat exchange performance and system efficiency. That is, a power plant using regenerative TFC with IHE is another motivating and promising alternative that should be studied in detail.

The optimization carried out in CHAPTER FIVE is simply rudimentary. The target function is the net work output of the heat recovery-to-power engine. A topic laudable of research is the optimization of working fluids for various heat sources and heat sink profiles. An 'auxiliary function' could be added to show the effects of the operating pressure, thermal conductivity and exergy density the working fluid.

Furthermore, the parametric study conducted in CHAPTER FIVE does not reflect specifically the effects of factors, like the condensing temperature and pressure, sizes of the component, and working fluid thermal conductivity, on the system efficiency and performance metrics. Nevertheless, because these are one of the vital factors in practical sense, a detailed systematic investigation of these parameters would be significant.

In order to avoid an environment susceptible to thermal pollution due to the condenser cooling fluid at fairly high temperature being dissipated as waste heat, an integrated heat recovery-to-power generation with process heating (for direct usage in a process) or district heating applications as a combined heat and power system is suggested. This can be an additional alternative to improve the system efficiency and energy utilization.

Lastly, after modelling and simulation, validating models simulation results by numerical and/ or experimental data is one of the challenges of novel modelling approaches. Though, modelling efforts are not always supported by the availability of experimental test rig, the setting up of test facilities are recommended for TFC engines particularly in the small- and medium-scale power range for further studies.

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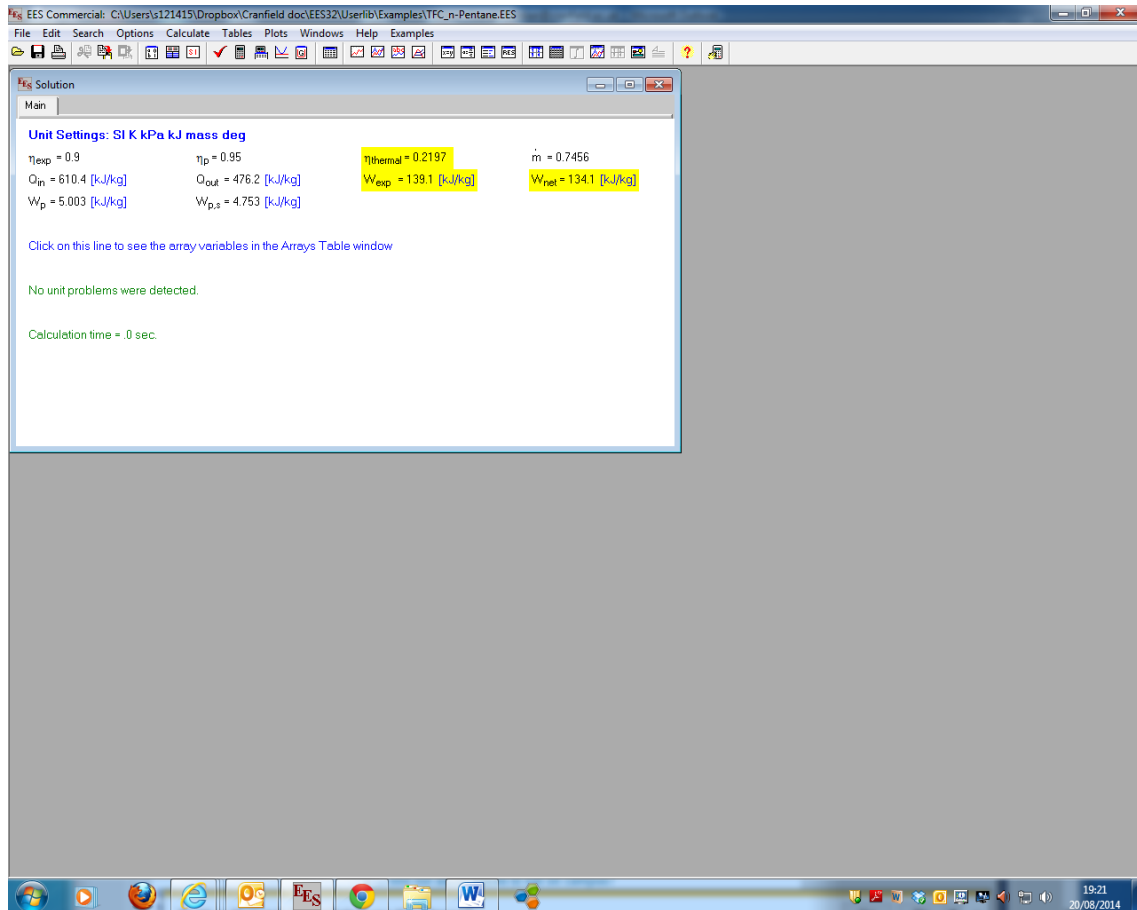
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## **APPENDICES**

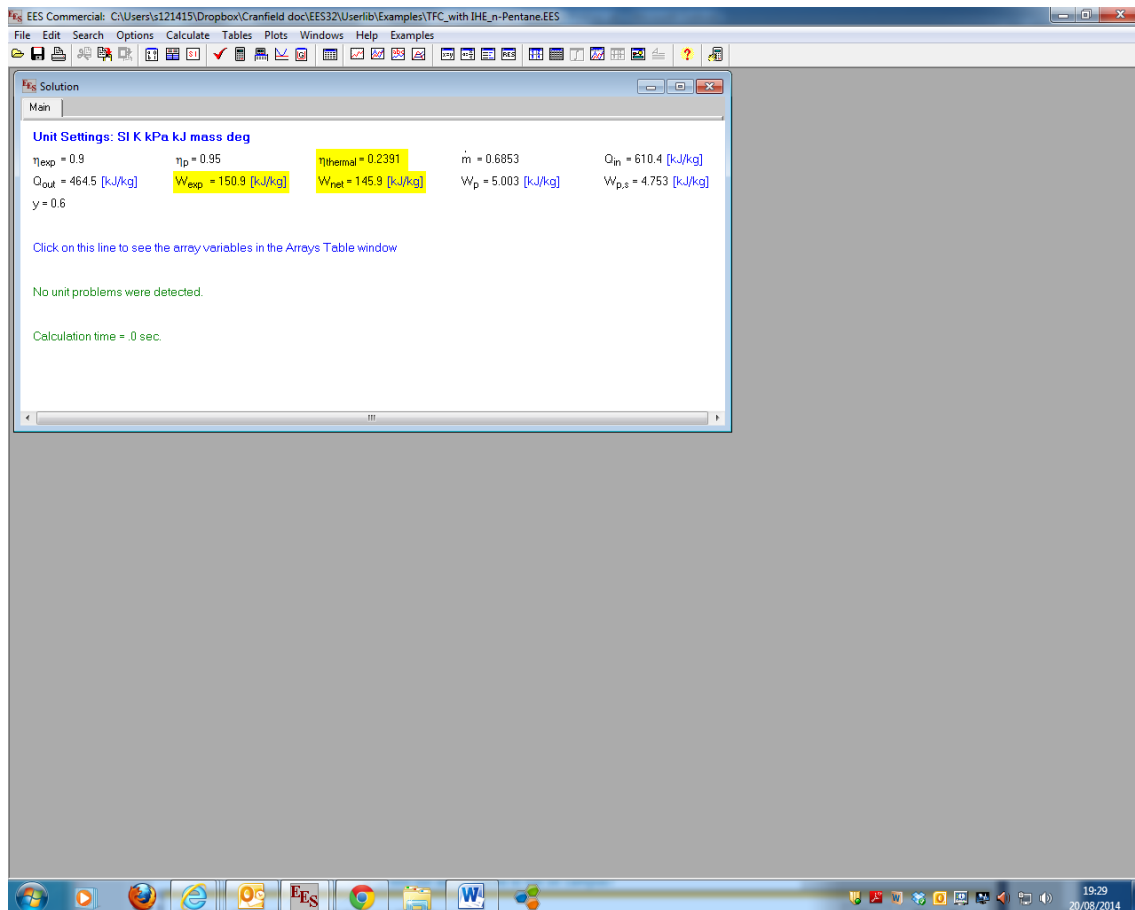
# Appendix A NUMERICAL MODELLING

## A.1 Simulation Results of Simple Trilateral Flash Cycle



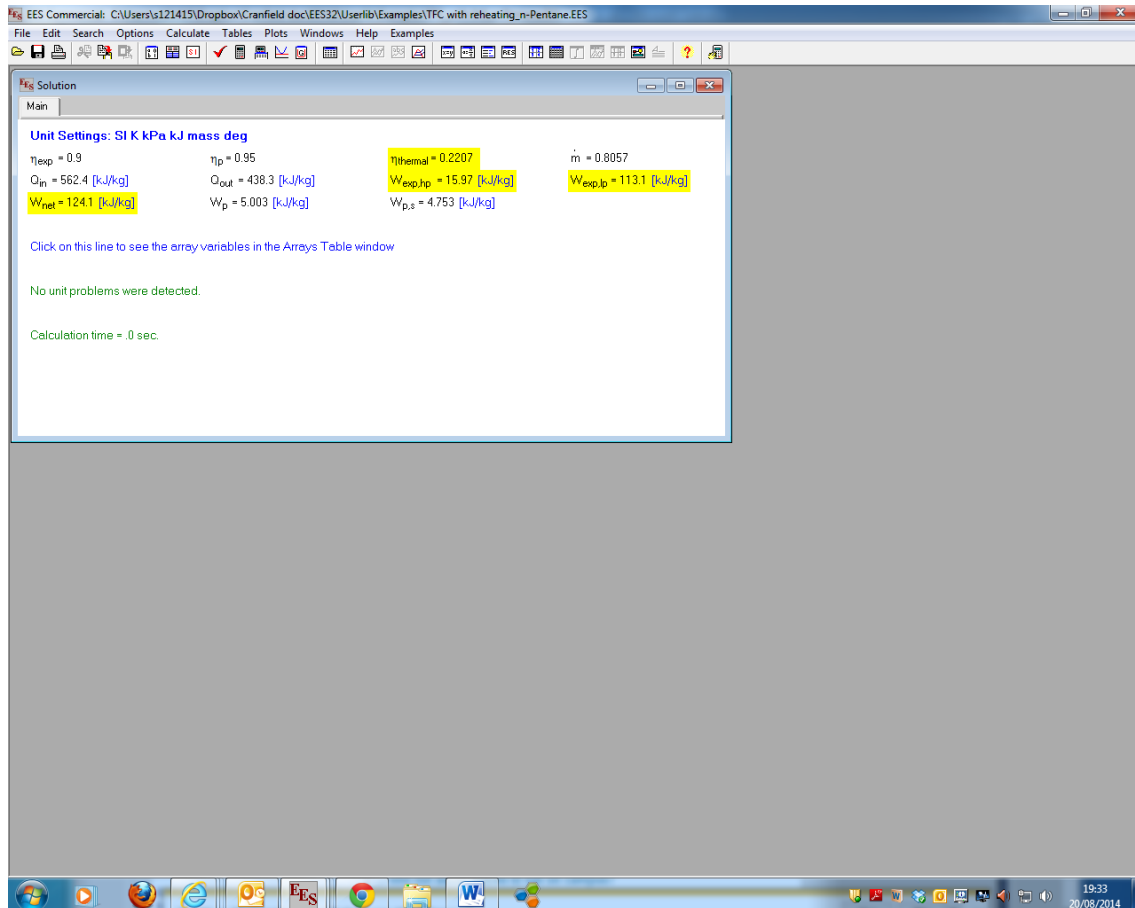
**Figure A.1:** Screen shot of the EES software displaying the simulation results of simple TFC

## A.2 Simulation Results of Trilateral Flash Cycle with Internal Heat Exchanger



**Figure A.2:** Screen shot of the EES software displaying the simulation results of TFC with IHE

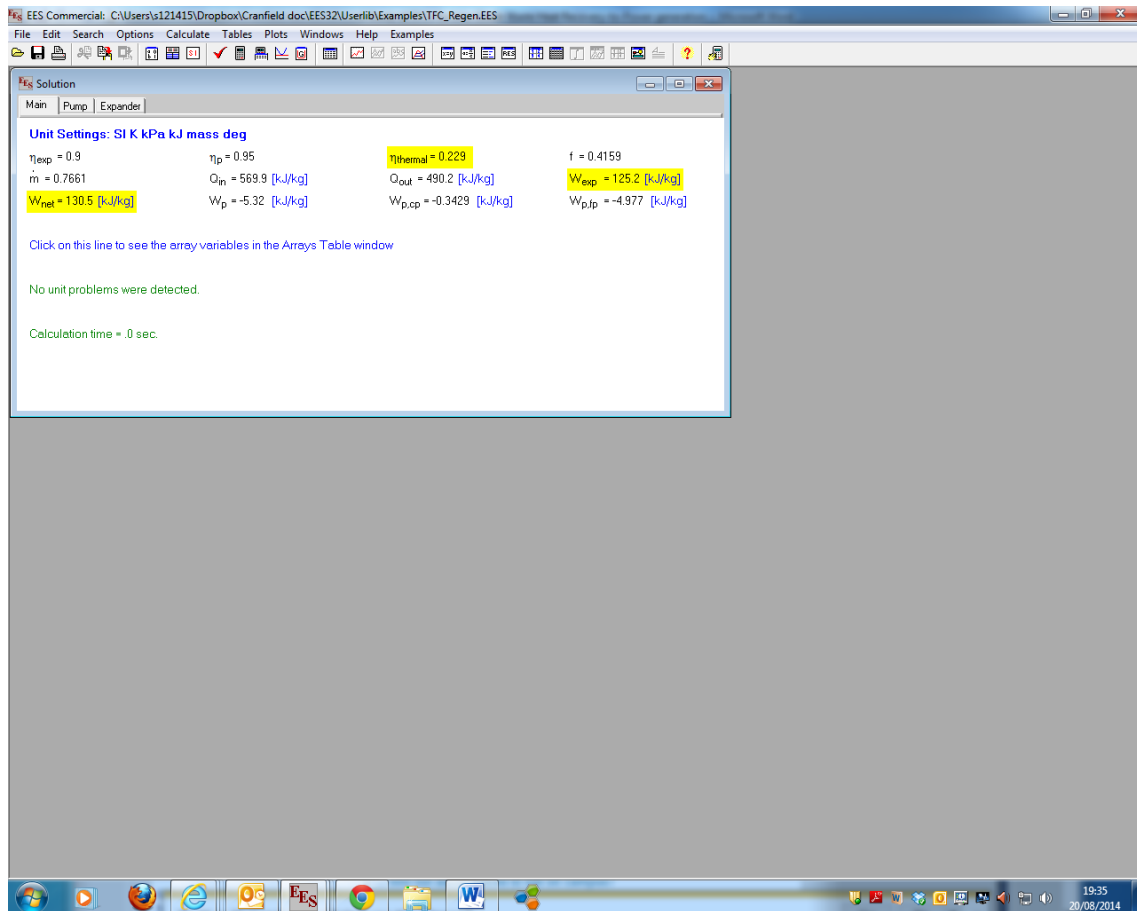
### A.3 Simulation Results of Reheat Trilateral Flash Cycle



**Figure A.3:** Screen shot of the EES software displaying the simulation results of reheat TFC



## A.4 Simulation Results of Regenerative Trilateral Flash Cycle



**Figure A.4:** Screen shot of the EES software displaying the simulation results of regenerative TFC

## Appendix B Cooling System

Assuming a water-cooling system is employed in the trilateral flash cycle power plants. The cooling water mass flow rate  $\dot{m}_{wat}$  in the cooling system at 297 K (point f) sub cooled to 293 K (point e) can be expressed as:

$$\dot{m}_{wat}(h_f - h_e) + \dot{m}_{wf}(h_4 - h_1) = 0 \quad (\text{B-1})$$

Which implies

$$\dot{m}_{wat} = \frac{\dot{m}_{wf}(h_4 - h_1)}{(h_f - h_e)} \quad (\text{B-2})$$

Putting  $\dot{m}_{wf} = 1 \text{ kgs}^{-1}$ ,  $h_4 = 499.6 \text{ kJkg}^{-1}$ ,  $h_1 = 23.36 \text{ kJkg}^{-1}$ ,  $h_f = 100.49 \text{ kJkg}^{-1}$ ,  $h_e = 83.34 \text{ kJkg}^{-1}$ ,  $s_f = 0.3510 \text{ kJkg}^{-1}\text{K}^{-1}$  and  $s_e = 0.2944 \text{ kJkg}^{-1}\text{K}^{-1}$  into Eq. (B-2), the cooling water mass flow rate  $\dot{m}_{wat}$  is computed to be  $27.77 \text{ kgs}^{-1}$ .